being placed between the wheels, and both driving the rear wheel. It will be noticed that the front crank-axle is connected by chain gearing to the rear crank-axle, the two axles rotating at the same speed ; the second chain passes over the larger wheel on the rear crank-axle and the chain-wheel of the driving-axle.

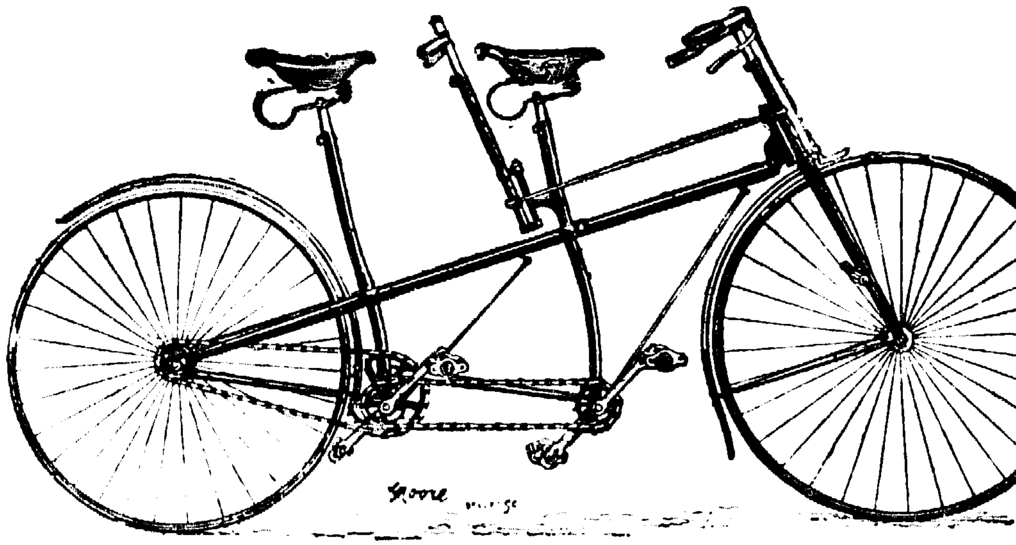


FIG. 139

Both riders have control of the steering, a light rod connecting the front fork to the rear steering-pillar. The long wheel-base of these bicycles adds to the steadiness of the steering at high speeds, since (see fig. 202), for the same deviation of the handle-bars, a machine with long wheel-base will move in a curve of larger radius than one with a shorter wheel-base. The distance between the wheel centres being much greater than in the single machine, the frame is subjected to very much greater straining actions, and imperfect design will be much more serious than in the single machine.

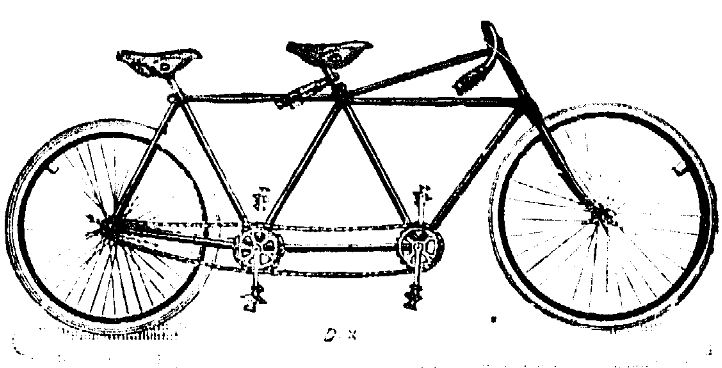


FIG. 140.

Figure 140 is an example of the present popular type of Tandem bicycle made by Messrs. Thomson and James. The machine is kinematically the same as that of figure 138, the particular difference being in the rear frame, which is of the diamond type, completely triangulated.

CHAPTER XV

DEVELOPMENT OF CYCLES: TRICYCLES, QUADRICYCLES, &c.

143. **Early Tricycles.**—No sooner was a practicable bicycle made than attention was turned to the three-wheeler as being the safer of the two machines, and offering some advantages, such as the possibility of sitting while the machine is at rest. It was very early found that the greater safety of the three-wheeler was more apparent than real. 'Velox,' writing in 1869, says, "Strange as it may appear to the uninitiated, the tricycle is far more likely to upset the tyro than the bicycle."

Figure 141 (from 'Velox's' book) represents a simple form of tricycle made in the sixties by Mr. Lisle, of Wolverhampton, known as the 'German' tricycle. It was, in fact, a converted 'Bone-shaker' bicycle, with the rear wheel removed and replaced by a pair of wheels running free on an axle two feet long. The motive power

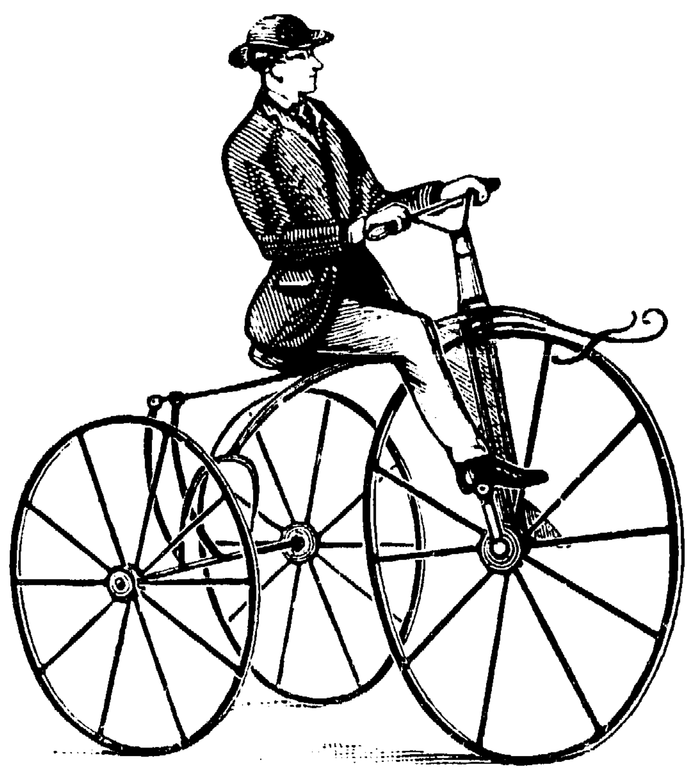


FIG. 141

was applied by pedals and cranks attached to the axle of the front wheel. A number of tricycles were made on the same general principle; but the weight of the rider being applied vertically over a point near the front corner of the wheel-base triangle, the margin of lateral stability was small. Mr. Lisle also made a ladies' double-

driving tricycle (fig. 142), in which the power was applied by treadles and levers acting on cranks on the axle of the rear wheels. Nothing is said about the axle of the rear wheels being divided,

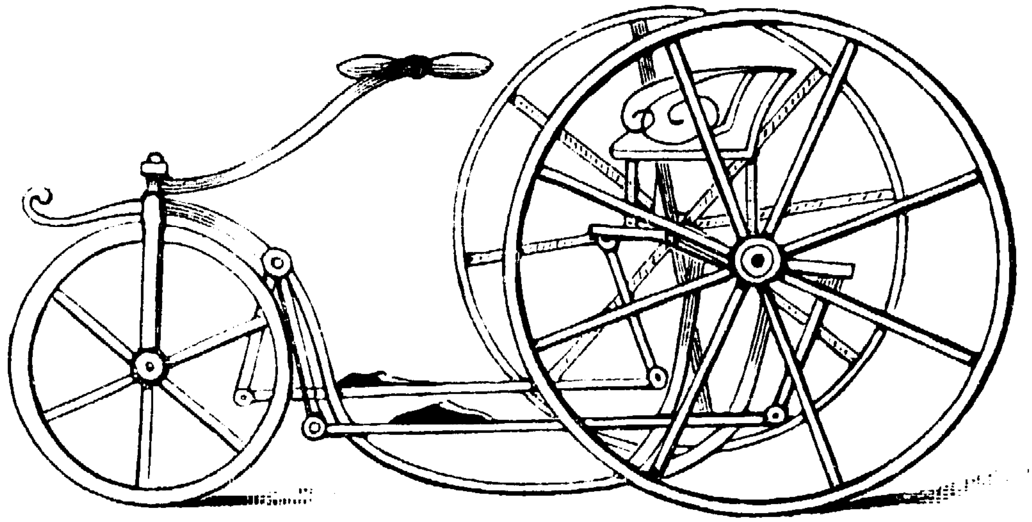


FIG. 142.

so it is probable that in turning round a corner the rear wheels skidded, just as is the case with railway rolling stock.

In the 'Dublin' tricycle (fig. 143) the driving-wheel was behind, and two steering-wheels placed in front; the margin of stability in case of a stoppage was much greater than in the 'German' tricycle (fig. 141). Another point of difference consisted in the application of the lever gearing; the pedals were fixed on oscillating levers, the motions of which were communicated by crank and connecting-rods to the driving-wheel.

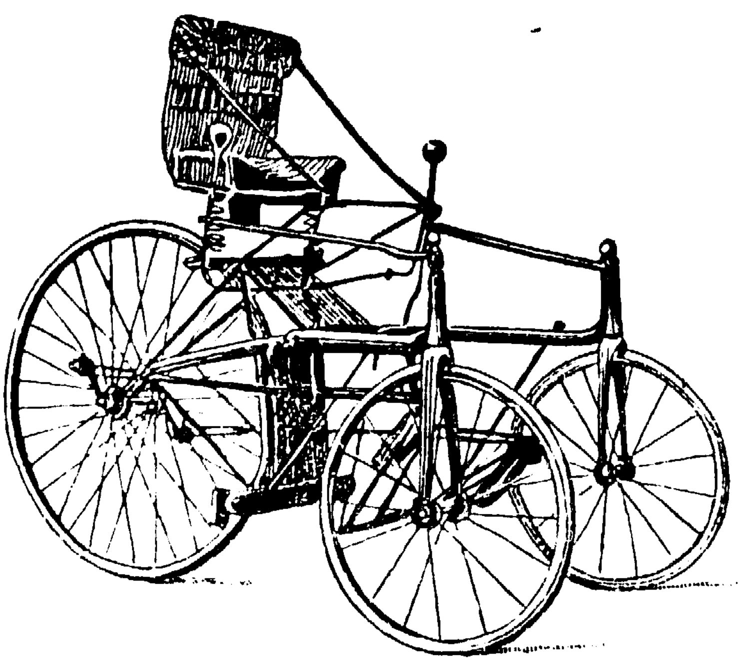


FIG. 143.

The 'Coventry' bicycle was at first made with lever gearing, but chain gearing was very soon afterwards applied to it. The 'Coventry Rotary' (fig. 144) was the most successful of the early

single-driving tricycles. It may be interesting to note that this type has been revived recently, the Princess of Wales having selected a tricycle of this type.

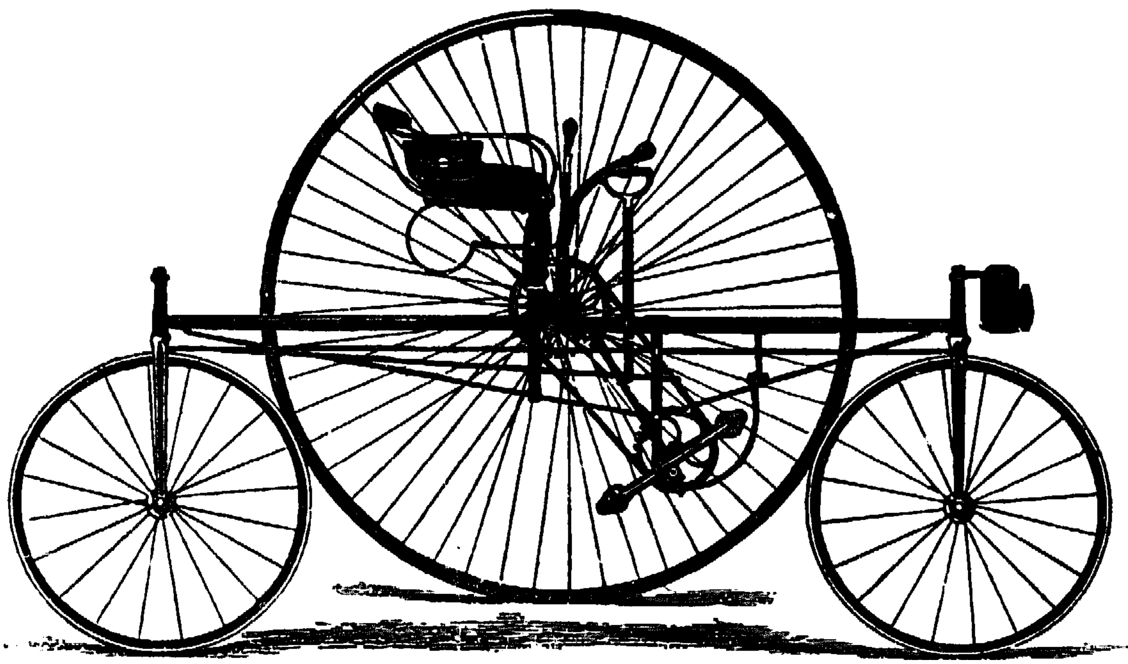


FIG. 144.

If the mass-centre be vertically over the centre of the wheel-base triangle, the pressure on each wheel will be one-third of the

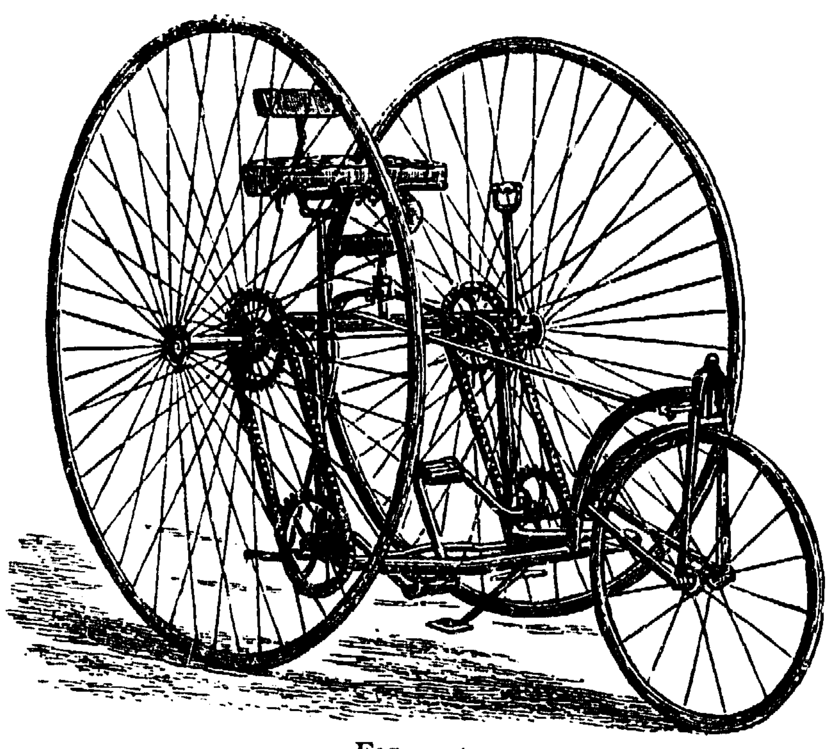


FIG. 145.

total weight. Under certain circumstances this pressure is insufficient for adhesion for driving, hence arose the necessity for

double-driving tricycles. In the 'Devon' tricycle, made in 1878 (fig. 145), which is fitted with chain gearing, the cog-wheels co-axial with the driving-wheels are fitted loose on their axles, and each cog-wheel drives its axle by means of a ratchet and pawl. In rounding a corner, the inside wheel is driven by the chain, while the outside wheel overruns its cog-wheel, the pawls of the ratchet-wheel being arranged so as to permit of this.

In the 'Club' tricycle (fig. 146), made by the Coventry Machinists Company in 1879, one of the wheels was thrown

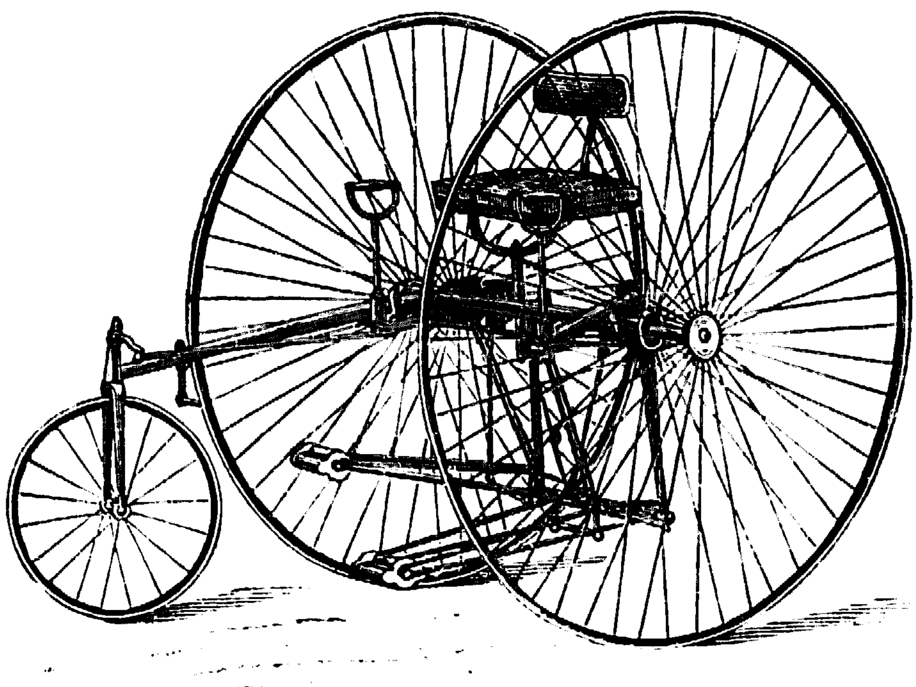


FIG. 146.

automatically out of gear when turning to one side or the other. Later, the same company used a clutch gear, somewhat similar in principle to the ratchet gear, but which had the advantage that the clutch could come into action at any point of the revolution, instead of only at as many points as there were teeth in the ratchet-wheel. The tricycle illustrated in figure 146 has only two tracks, which, in the early days of tricycles, was supposed to be of some advantage, in so far that it was easier to pick out two good portions along a bad piece of road than three.

A number of single and side-driving, rear-steering tricycles (fig. 147) were made about the years 1879 and 1880, but on account of their imperfect steering they were sometimes found extremely dangerous, and their manufacture was soon abandoned

in favour of double-driving rear-steerers, of which the 'Cheylesmore' (fig. 148), made by the Coventry Machinists Company, was

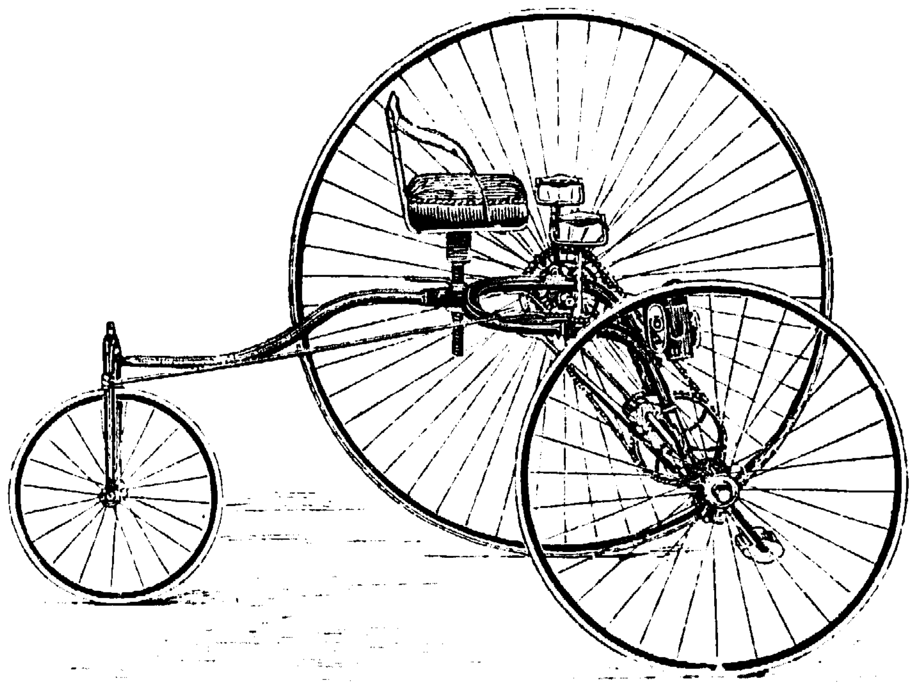


FIG. 147.

one of the most successful. Tradesmen's carrier tricycles are still made of this type.

144. **Tricycles with Differential Gear.**—The front-steering, double-driving tricycle with loop frame, as in figure 145, next

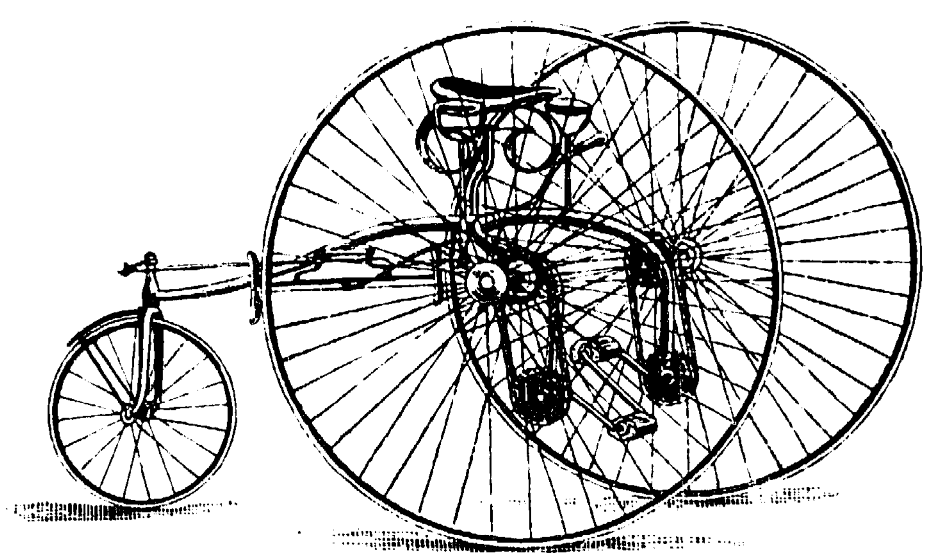


FIG. 148.

became more and more popular. The invention by Mr. Starley of the 'Differential' tricycle axle or *balance-gear* marks a great step in the development of the three-wheeler. This gear, or its

equivalent, has been ever since used for double-drivers, clutch and ratchet gears having been abandoned.

As improvements in detail were slowly introduced, the lever gear fell into disuse (which is easily accounted for by the fact that with it gearing either up or down is impossible), and chain gearing became universal. With chain gear, and the possibility of gearing up, the driving-wheels were made gradually smaller and smaller, with a consequent reduction in the weight of the machine.

The 'Humber' tricycle met with great success on the racing path, but, on account of its tendency to swerve on passing over a

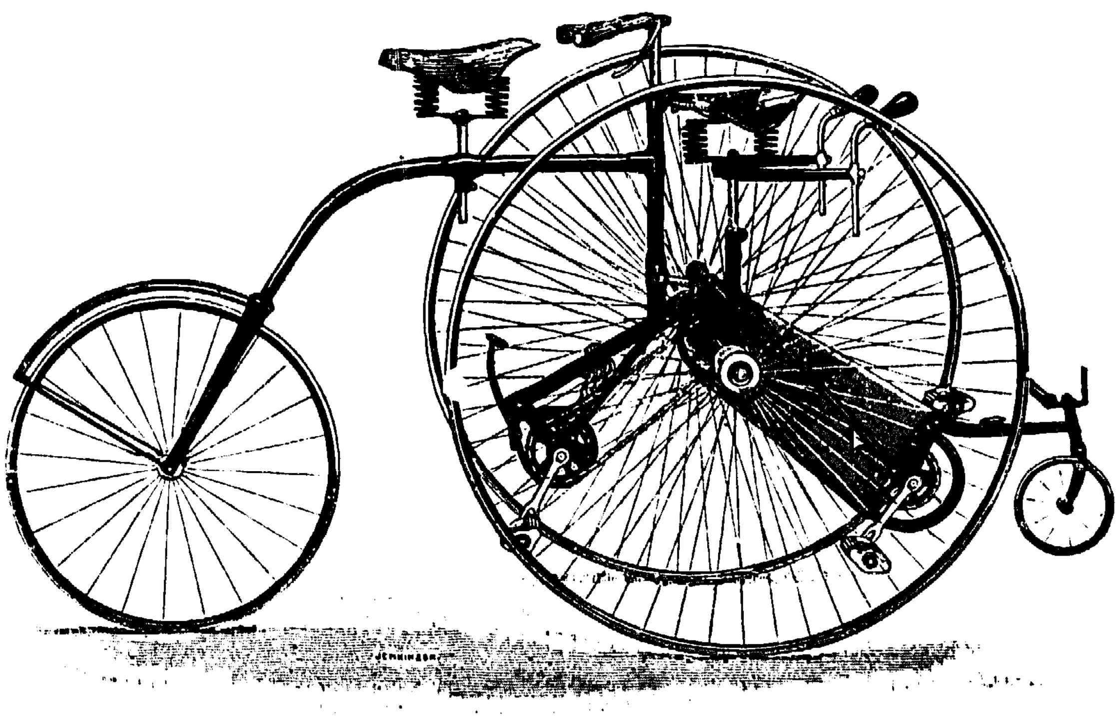


FIG. 149.

stone, its success as a roadster was not so marked. When used as a tandem (fig. 149), with one rider seated on the front-frame supporting the driving-axle, the tendency to swerve was reduced and the safety increased (see sec. 183). In a later type this difficulty was overcome by converting the machine into a rear-steerer, the steering-pillar being connected by light levers and rods to the steering-wheel.

The loop-frame tricycle was gradually superseded by one with a central frame, in which the steering-wheel was actuated direct by the handle-bar, the result being the 'Cripper' tricycle (fig. 150). In this, as made by Messrs. Humber & Co., the chain lies

in the same plane as the backbone; the crank-bracket being suspended from the backbone and the gear being exactly central.

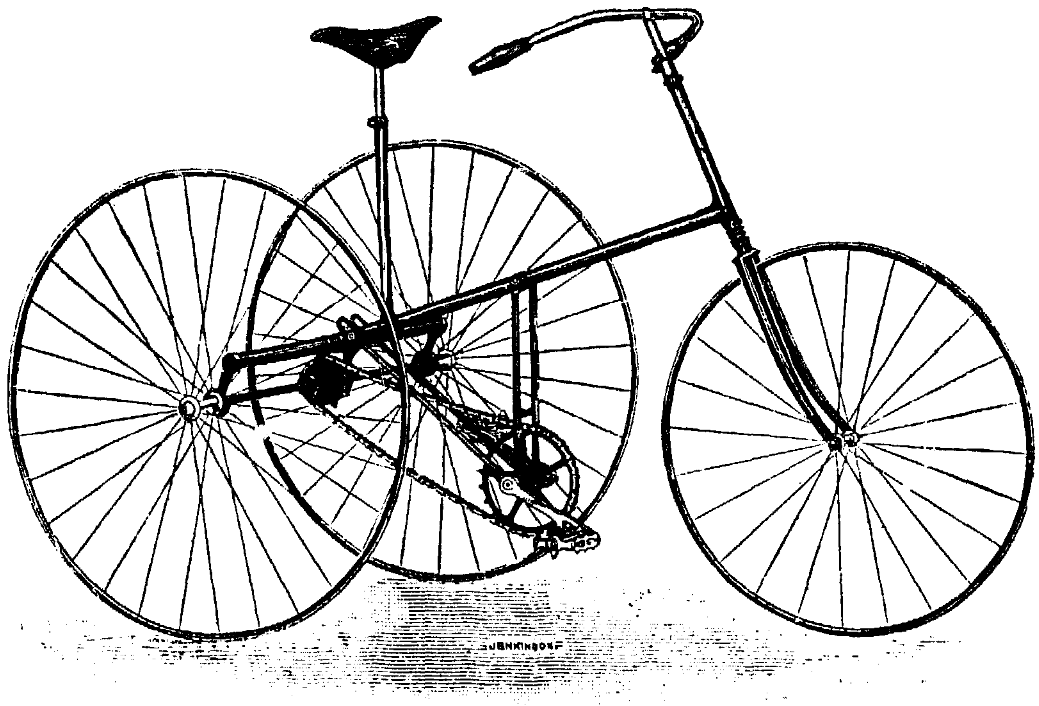


FIG. 150.

The axle is supported by four bearings, though the axle-bridge, with four bearings, had already been used in the 'Humber' tricycles.

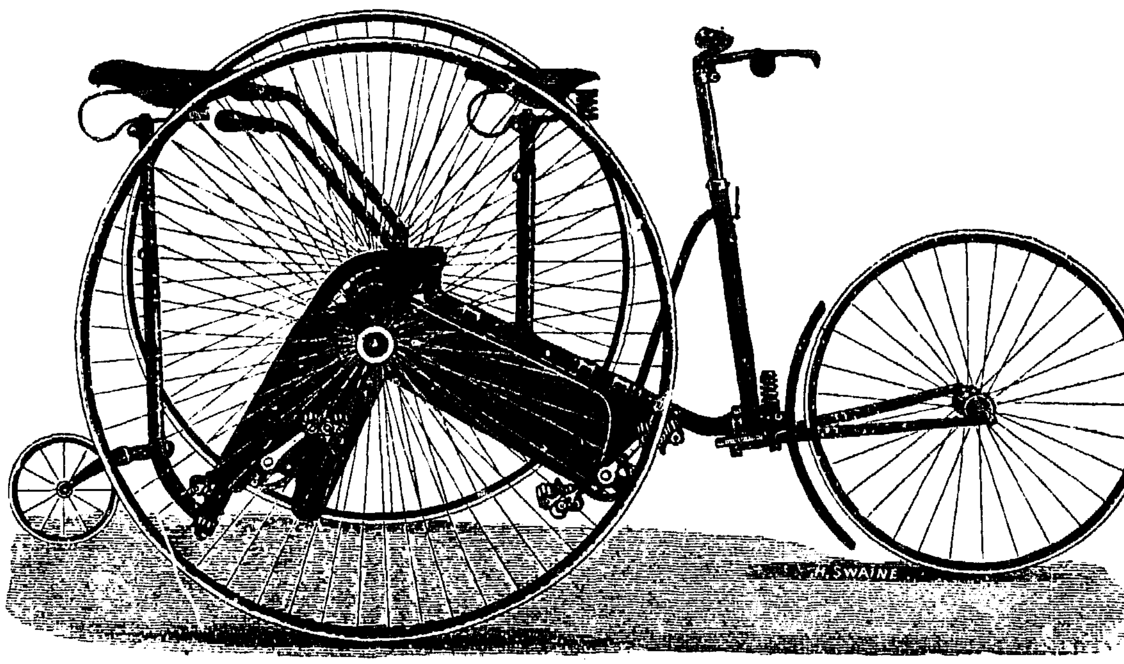


FIG. 151.

Among the successful tricycles of this period may be mentioned the 'Quadrant,' in which the steering-wheel was not

mounted in a fork, but the ends of the spindle ran on guides in the frame (sec fig. 254), and the 'Rudge Royal Crescent' (fig. 151), in which the fork of the steering-wheel was horizontal, and the steering-axis intersected the ground some considerable distance between the point of contact of the steering-wheel.

Up to the year 1886 the 'Ordinary' bicycle had a very great influence on tricycle design, the driving-wheels of tricycles being usually made very large (in fact, sometimes they were geared *down* instead of up) and the steering-wheel small. The weight of two large wheels was a serious drawback, while the excessive vibration from the small steering-wheel was a source of great

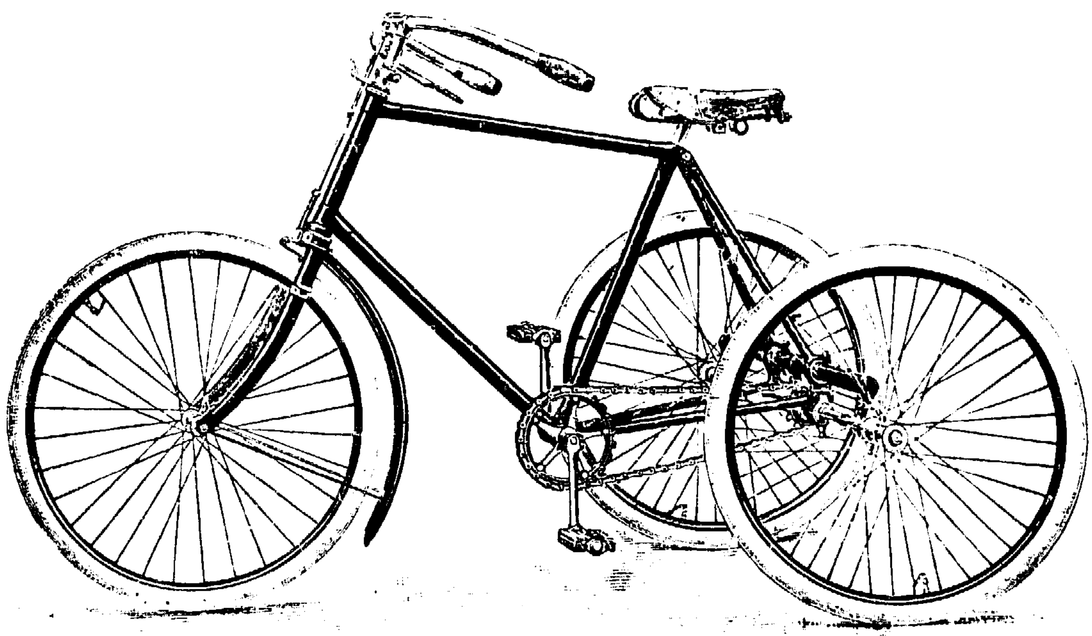


FIG. 152.

discomfort to the rider. The distance between the wheel centres was usually made as small as possible, the idea being that the tricycle should occupy little space. Common measurements for 'Cripper' tricycles at this time were: Driving-wheels, 40 in. diam.; steering-wheel, 18 in. diam.; distance between driving- and steering-wheel centres, 32 in.; driving-wheel tracks, 32 in. apart. Weight: Racers, 40 lbs.; roadsters, 70–80 lbs.

The size of the driving-wheel has been gradually diminished, that of the steering-wheel increased, until now (1896) 28 in. may be taken as the average value for the diameter of each of the three wheels. The wheel centres have been put further apart,

42–45 in. being now the usual distance, the comfort of the rider and the steadiness of steering being both increased thereby.

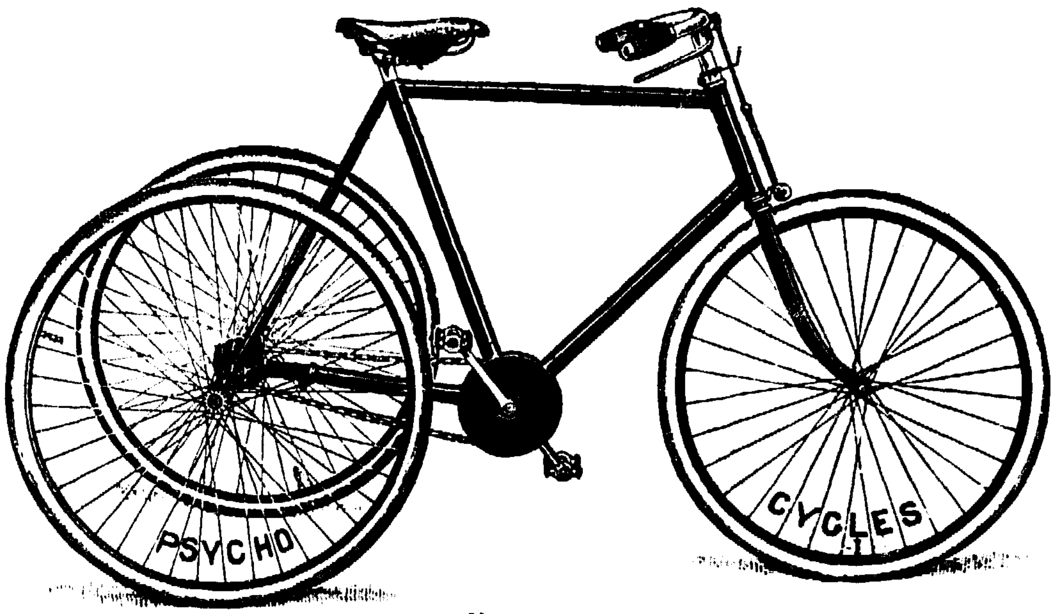


FIG. 153.

The design of frame has also been greatly improved, so that the weight of a roadster has been reduced to 40–45 lbs. without in any way sacrificing strength.

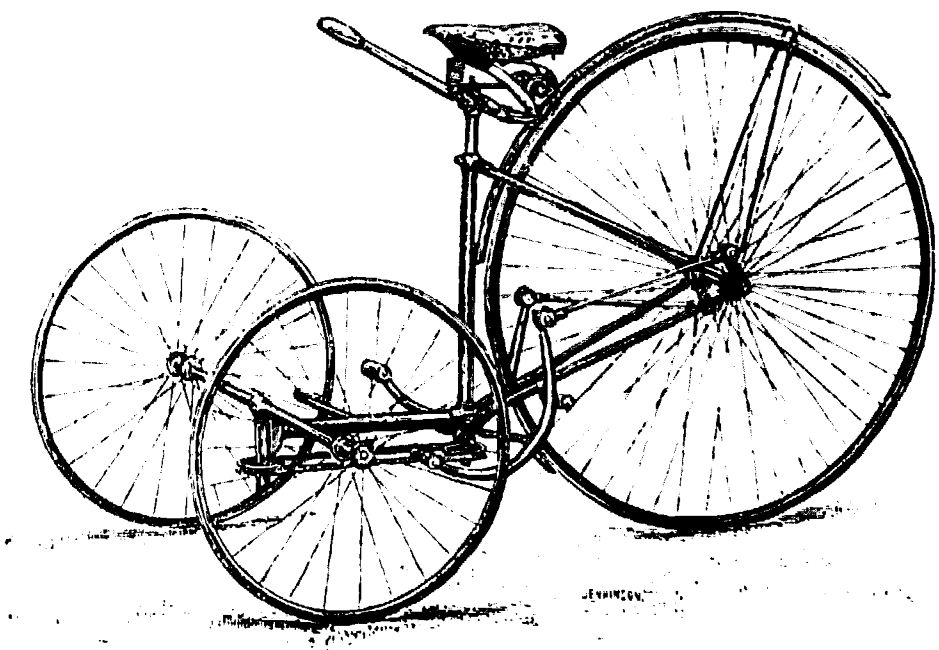


FIG. 154.

Figure 152 shows a tricycle by the Premier Cycle Company, Ltd., embodying these improvements. The frame and chain gearing is almost identical with that of the bicycle; the balance-gear and axle-bridge, with its four bearings, being added.

Figure 153 shows a tricycle by Messrs. Starley Bros., in which the bridge is a tube surrounding, and concentric with, the axle,

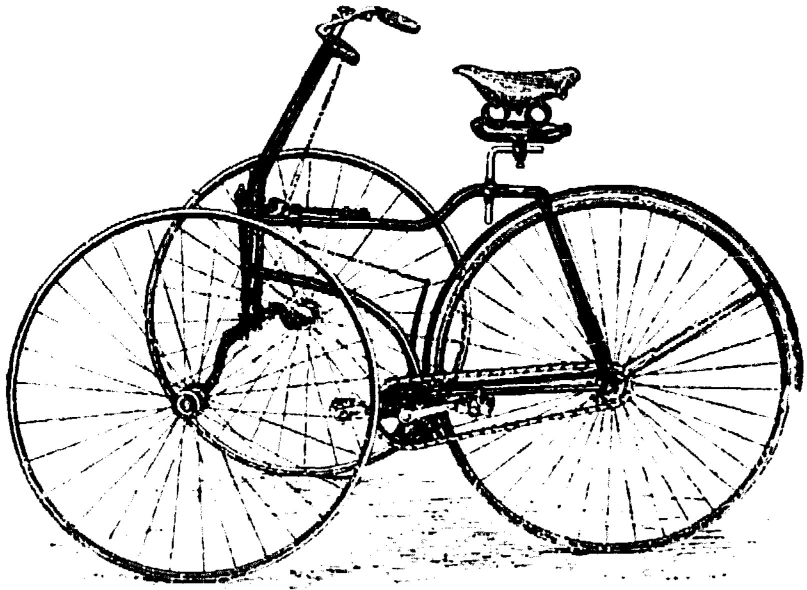


FIG. 155.

and the gear is exactly central; so that the frame is considerably simplified, and the appearance of the machine vastly improved.

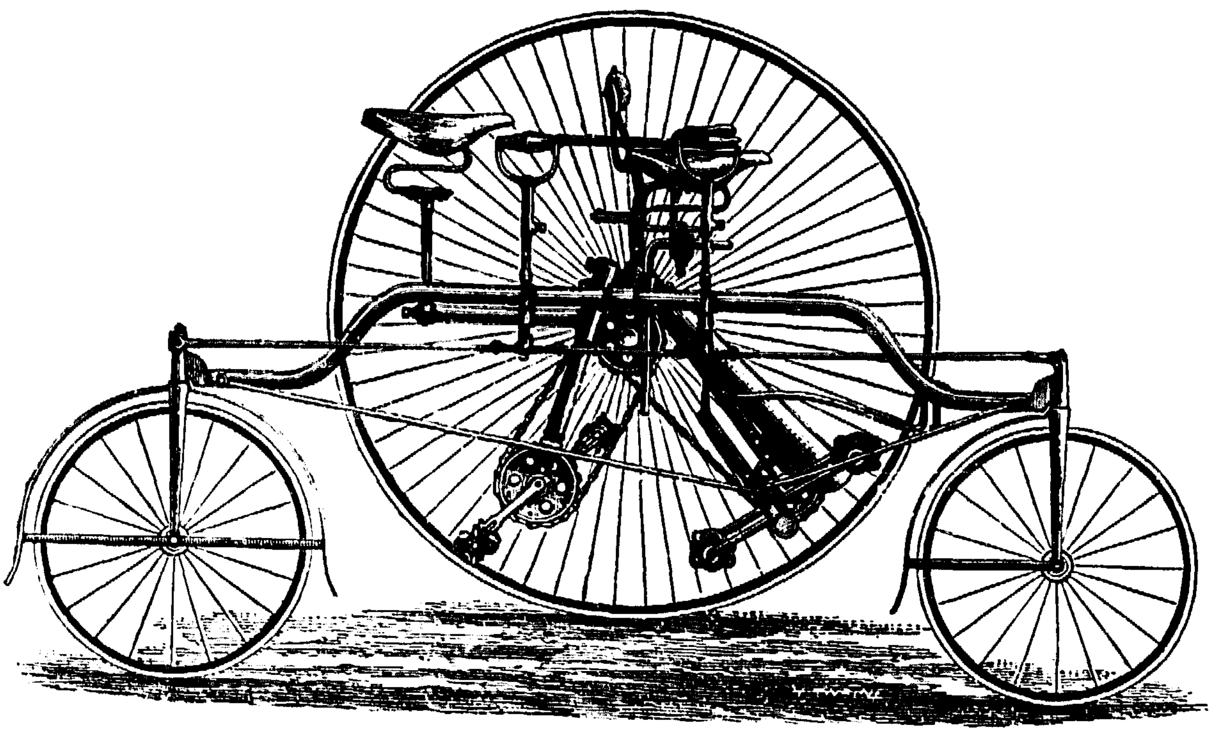


FIG. 156.

This may be taken as the highest point reached in the development of the 'Cripper' type of tricycle.

145. **Modern Single-driving Tricycles.**—Several successful

single-driving rear-driver tricycles have been made, among them being the 'Facile Rear-Driver' (fig. 154) and the 'Phantom'

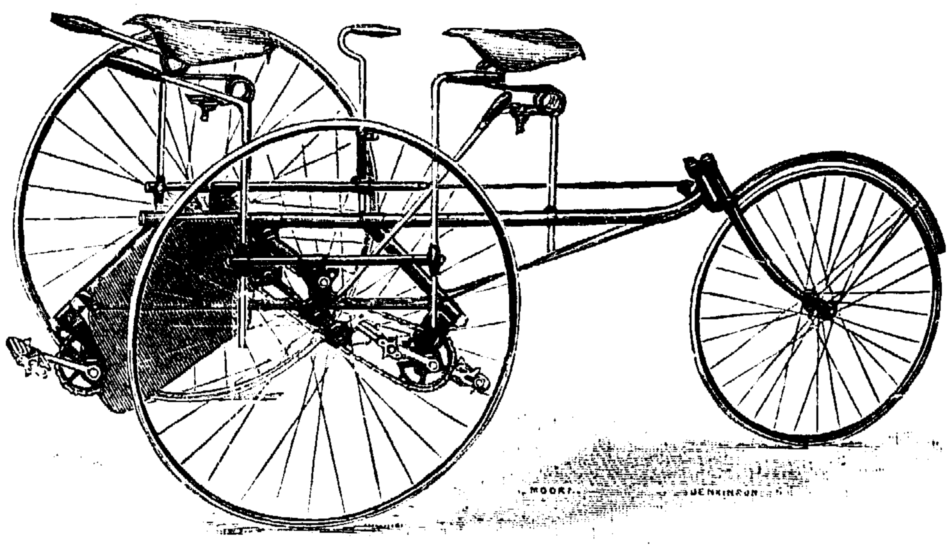


FIG. 157.

(fig. 155). In these the two idle (or non-driving) wheels run freely on an axle supported by the front frame. These tricycles

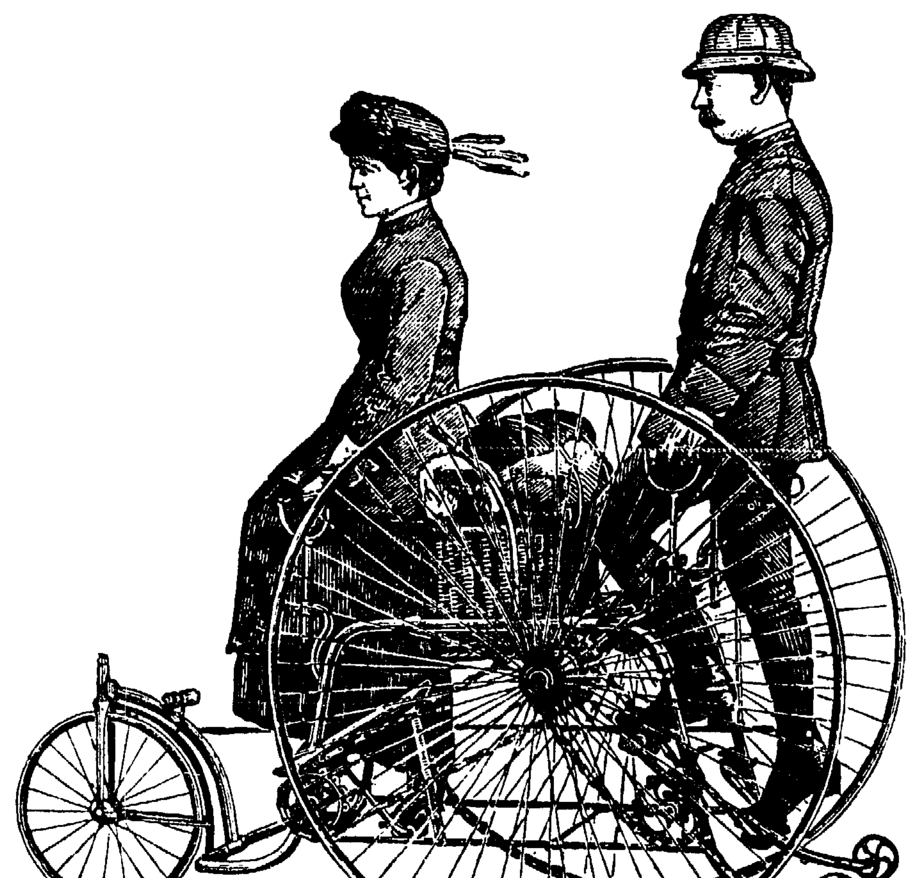


FIG. 158.

are subject to the same faults of swerving as the 'Humber' tricycle.

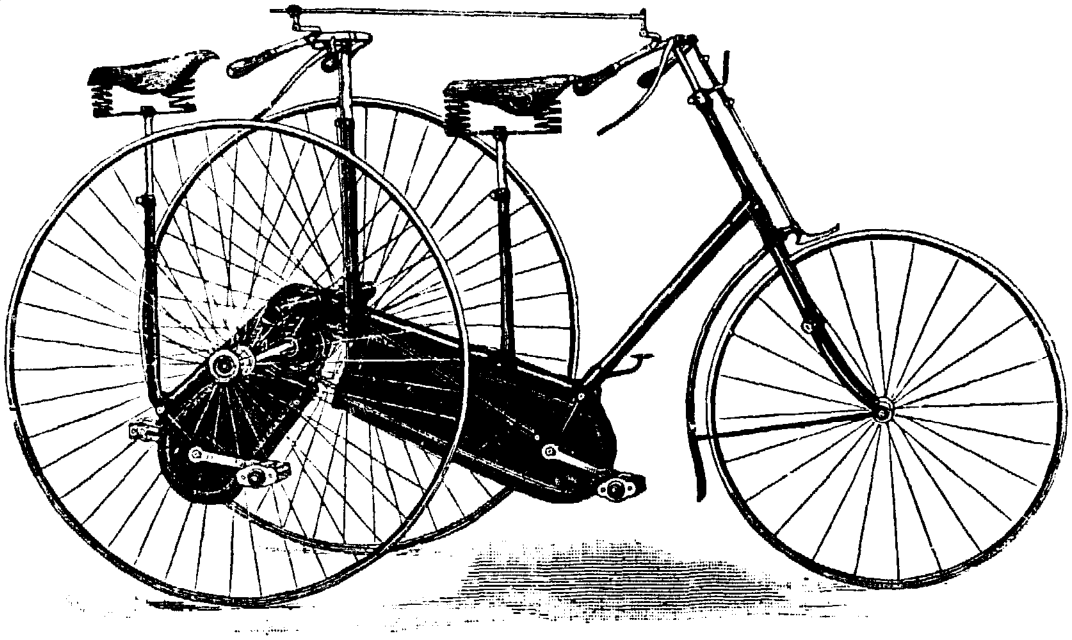


FIG. 159.

An important improvement is effected by mounting each wheel on a short axle, which can turn about a vertical steering-

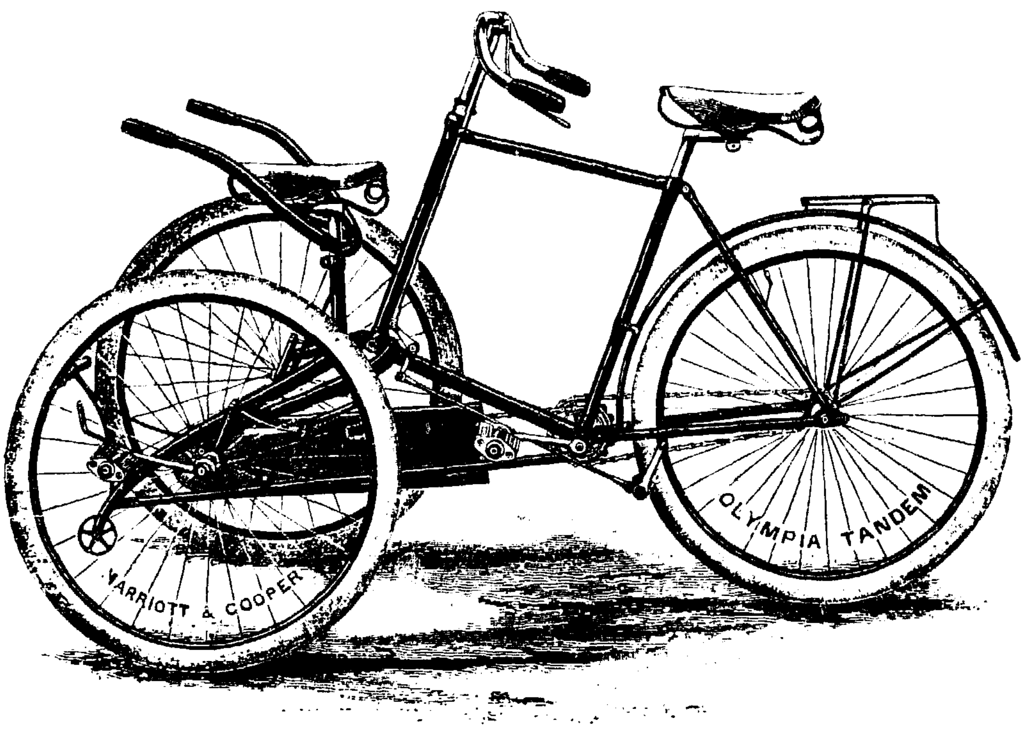


FIG. 160

head placed as close as possible to the wheel, as in the 'Olympia' (fig. 160), one of the most successful of modern tricycles.

146. **Tandem Tricycles.**—Tricycles for two riders were soon brought to a relatively high state of perfection, and were almost,

if not quite, as popular as tricycles for single riders. Among the earliest may be mentioned the 'Rudge Coventry Rotary' (fig. 156),

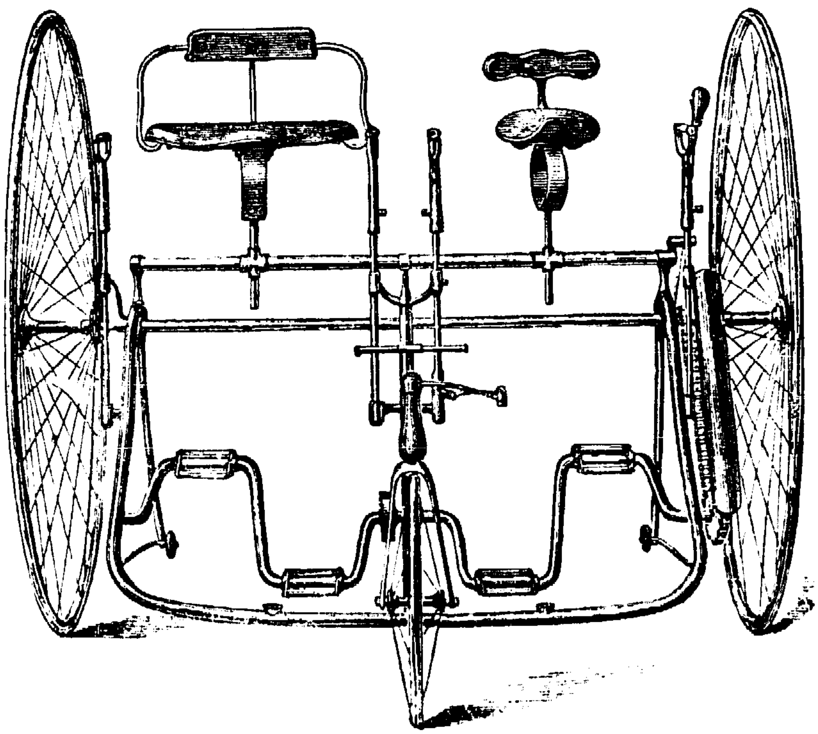


FIG. 161.

the 'Humber' (fig. 149), the 'Invincible' rear-steerer (fig. 157), and the 'Centaur' (fig. 158). Later, the 'Cripper' (fig. 159) and the 'Royal

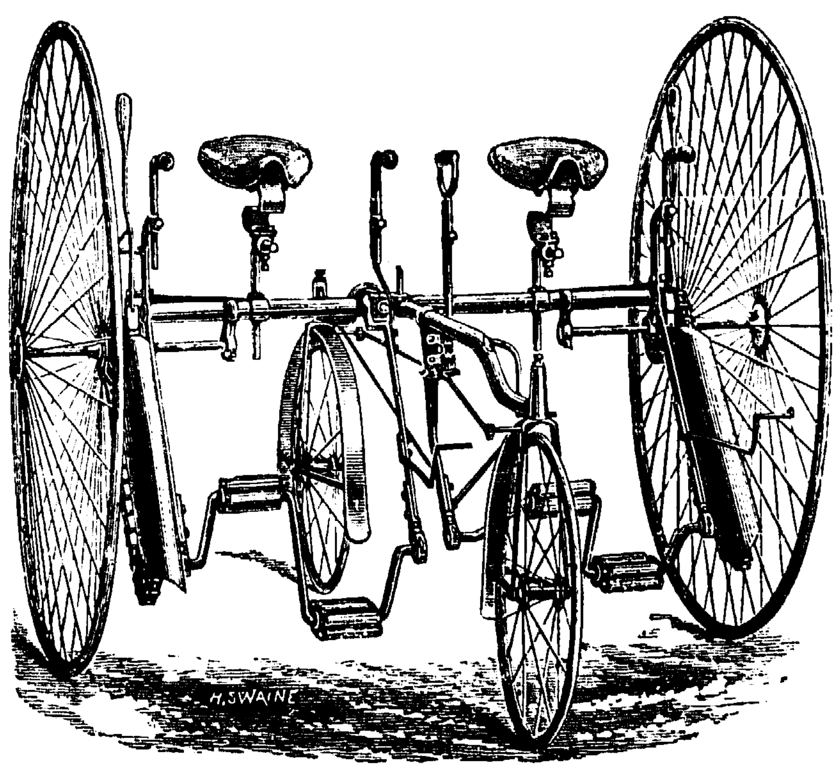


FIG. 162.

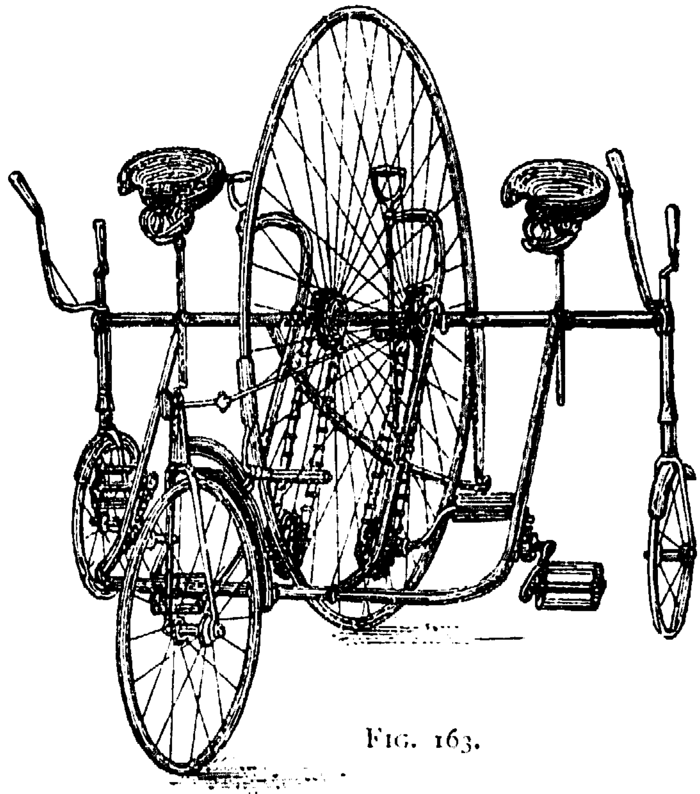


FIG. 163.

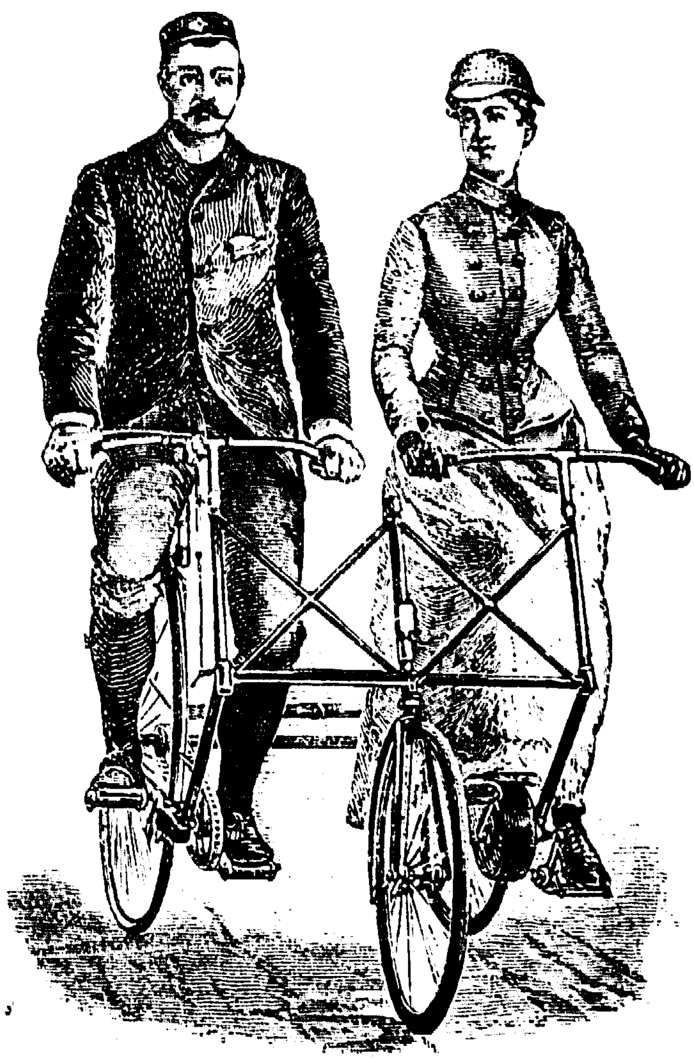


FIG. 164.

Crescent' (fig. 151) were made as tandems. In all these tandems, with the exception of the 'Coventry Rotary,' one of the riders overhangs the wheel-base, so that the load on the steering-wheel is actually less than when a single rider used the machine. The 'Coventry Rotary' is a single-driver, the others are double-drivers.

The most successful modern tandem tricycle is the 'Olympia' (fig. 160), a single-driver.

147. **Sociables,** or tricycles for two riders sitting side by side, were at one time comparatively popular. Figure 161 shows one with a loop frame made by Messrs. Rudge & Co., which, by the removal of certain parts, could be converted into a single tricycle; figure 162, a 'Sociable' formed by adding another driving-wheel, crank-axle, and seat to the 'Coventry Rotary' (fig. 144).

In the 'One-track Sociable' (fig. 163),

made in 1886 by Mr. J. S. Warman, the weight of the rider rested mainly on the two central wheels, the small side wheels merely preventing the machine overturning when starting and stopping. It was, in fact, a sociable bicycle with two side safety-wheels added.

In the 'Nottingham Sociable' tricycle (fig. 164), made by the Nottingham Cycle Co. in 1889, each rider sat directly over the

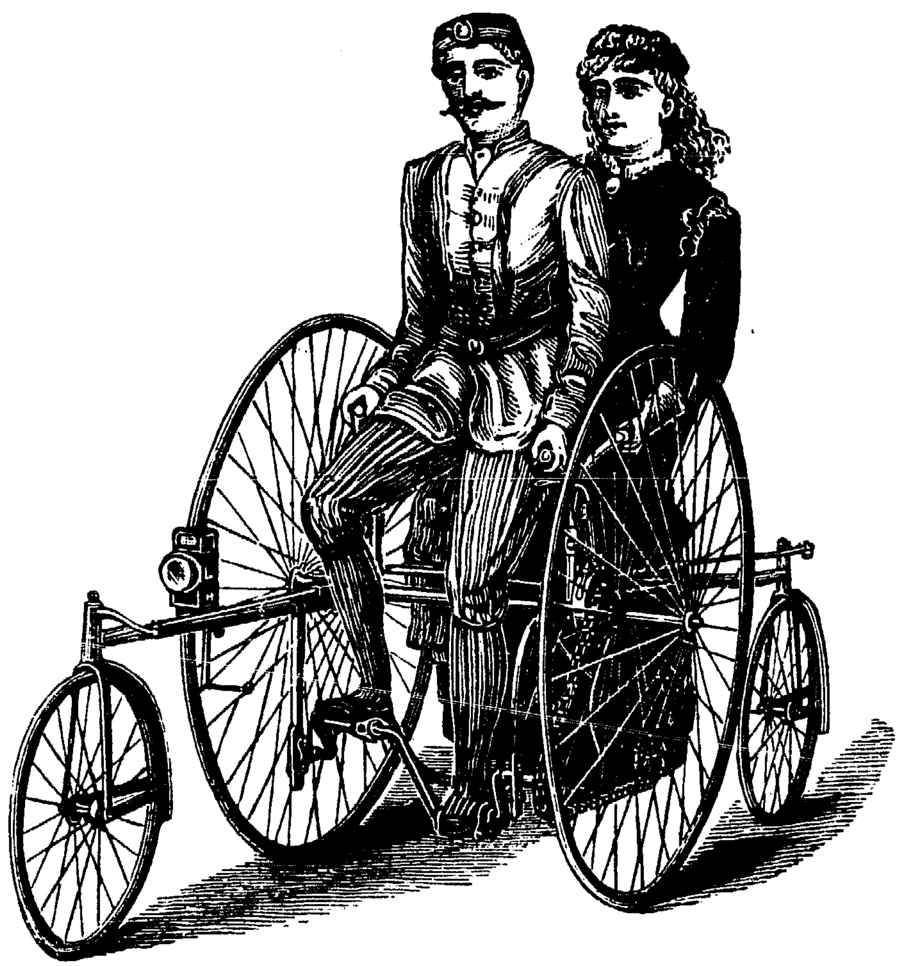


FIG. 165.

rear portion of a 'Safety' bicycle, and the heads of the two frames were united by a trussed bridge to a central steering-head.

148. **Convertible Tricycles.**—A great many machines for two riders were at one time made by adding a piece to a tricycle so as to form a four-wheeler. Of these *convertible tricycles*, as they were called, the 'Royal Mail' two-track machine (fig. 165) and the 'Coventry Rotary Sociable' (fig. 162) may be noticed.

Figure 166 shows the 'Regent' tandem tricycle, formed by

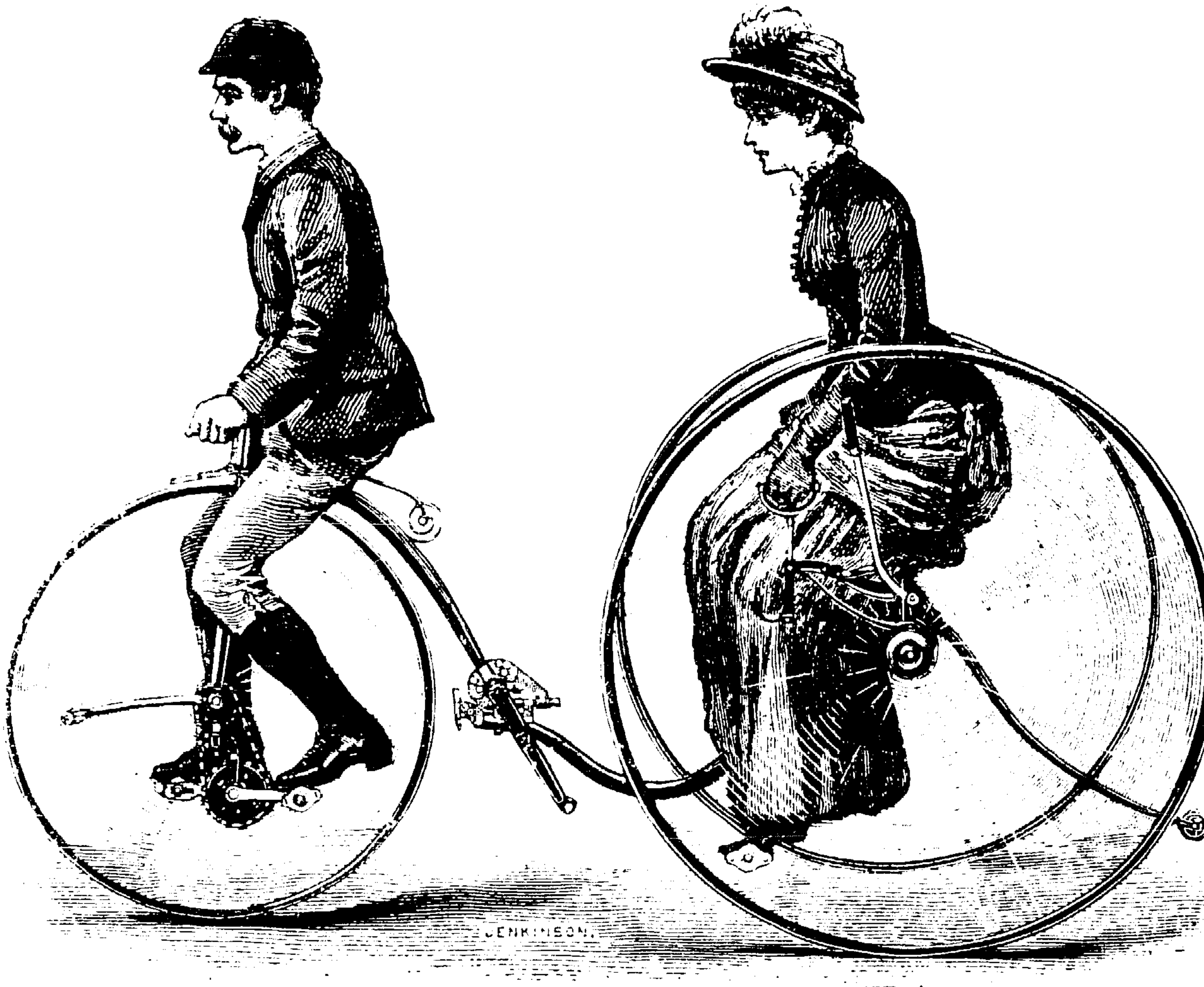


FIG. 166.

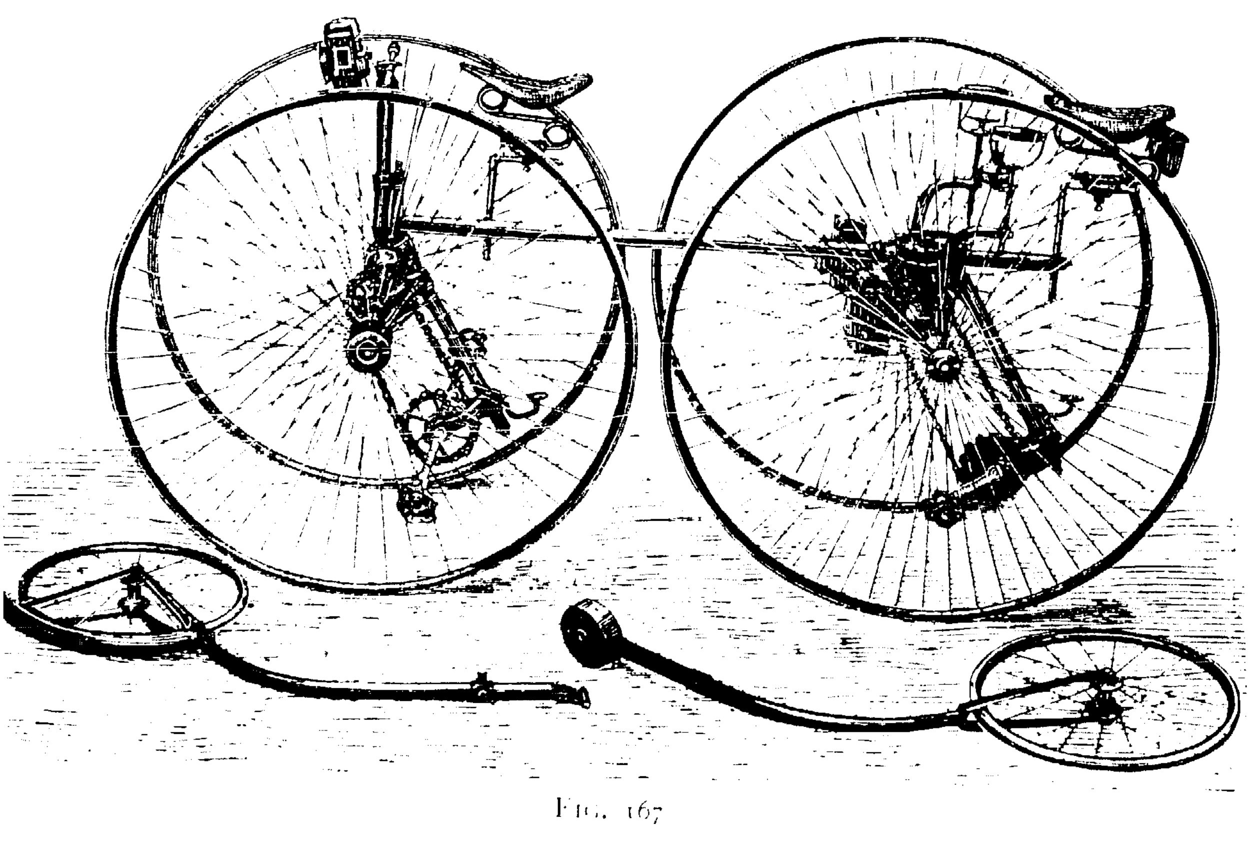


FIG. 167

coupling the front wheel and backbone of a 'Kangaroo' bicycle to the rear portion of a 'Cripper' tricycle; affording an example of a *treble-driving* cycle.

Figure 167 shows a four-wheeler formed by coupling together the driving portions of a 'Humber' and a 'Cruiser' tricycle, affording an example of a *quadruple-driving* cycle, all four wheels being used as drivers.

149. **Quadricycles.**—With the exception of the convertible tricycles above referred to, comparatively few four-wheeled cycles have been made. In 1869 'Velox' wrote: "No description of velocipedes would be perfect without some allusion to the favourite

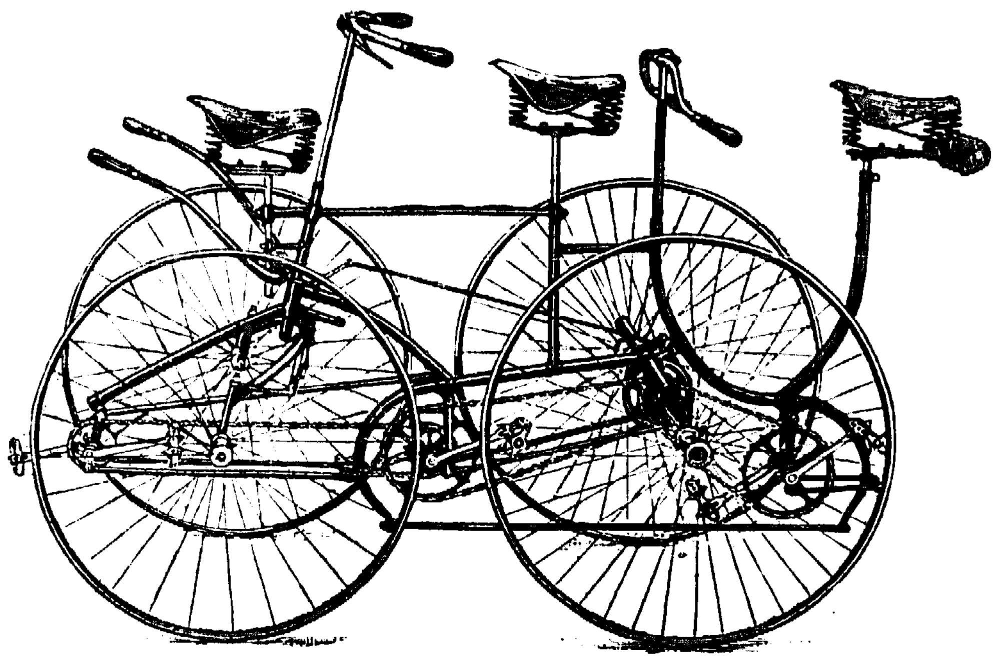


FIG. 168.

our-wheeler of the past generation of mechanics." Figures 117 and 118 show one of the best as manufactured by Mr. Andrews, of Dublin. The frame was made of the best inch square iron 7 feet long between perpendiculars, and was nominally rigid, so that in passing over uneven ground either the frame was severely strained or only three wheels touched the ground. The two driving-wheels were fixed at the ends of a double cranked-axle driven by lever gear, the path of each pedal being an oval curve with its longer axis horizontal. While moving in a circle, the driving-wheels skidded as well as rolled, since the outer had to move over a greater distance than the inner.

Bicycles and tricycles have almost monopolised the attention of cycle makers, and no practicable quadricycle was made until Messrs. Rudge & Co. produced their 'Triplet' quadricycle (fig. 168) in 1888. The front-frame supporting the two side steering-wheels can swing transversely to the rear-frame, so that the four wheels always touch the ground, however uneven, without straining the frame. The same design was applied to a quadricycle for a single rider.

CHAPTER XVI

CLASSIFICATION OF CYCLES

150. **Stable and Unstable Equilibrium.**—Cycles may be divided into two great classes, according as the static equilibrium during the riding is stable or unstable. The former class may be

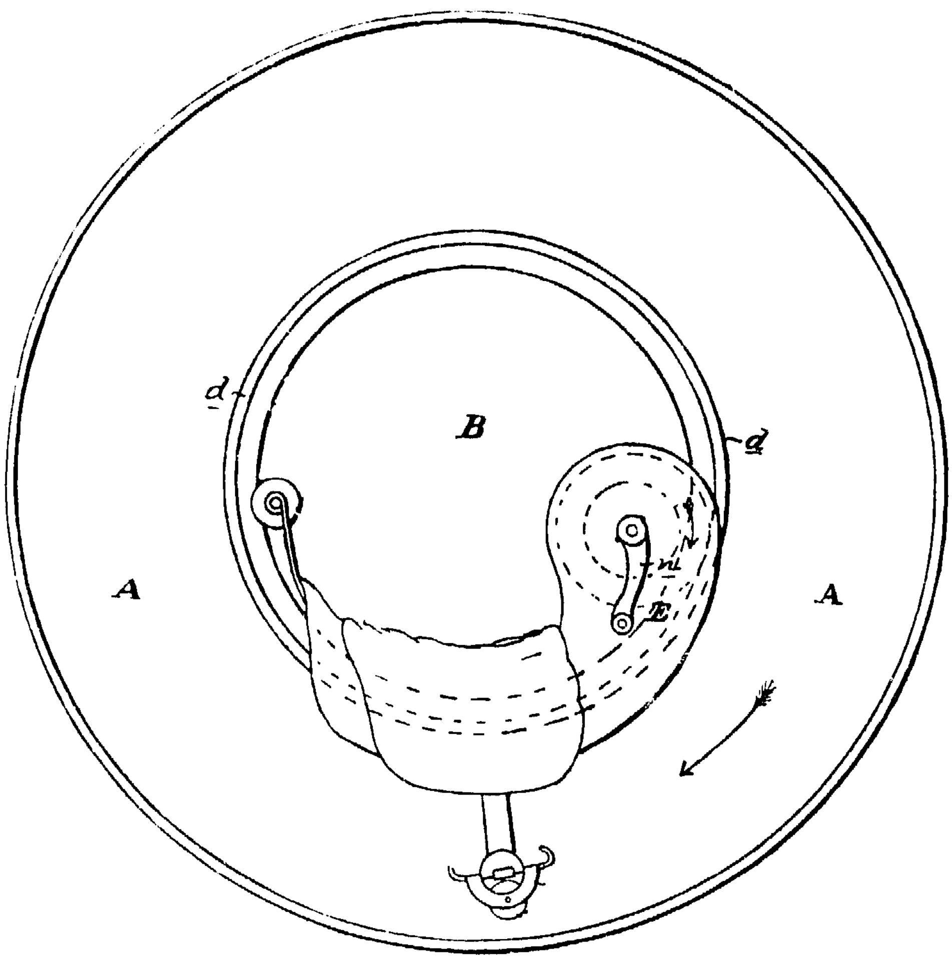


FIG. 169.

further separated into three divisions : (*a*) Tricycles, in which the frame, supported as it is at three points, is a statically determinate structure ; (*b*) Multicycles, having four or more wheels, the frame

generally having a hinge or universal joint, so that the wheels may adjust themselves to any inequalities of the ground. If the frame be absolutely rigid it will be a statically indeterminate structure. (*c*) Dicycles of the 'Otto' type, with two wheels, in which the mass-centre of the machine and rider is lower than the axle. No machine of this class has ever been made, to the author's knowledge.

Cycles with unstable equilibrium may be divided into three classes, according to the direction in which the unstable equilibrium exists: *Monocycles*, having only one wheel; *Bicycles*, having two wheels forming one track; and

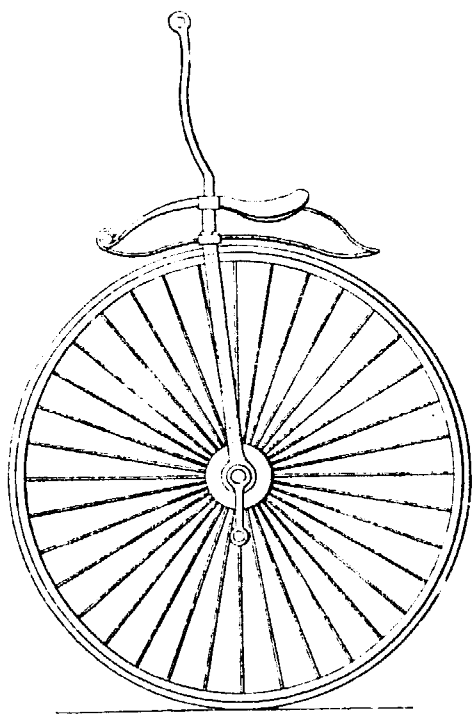


FIG. 170.

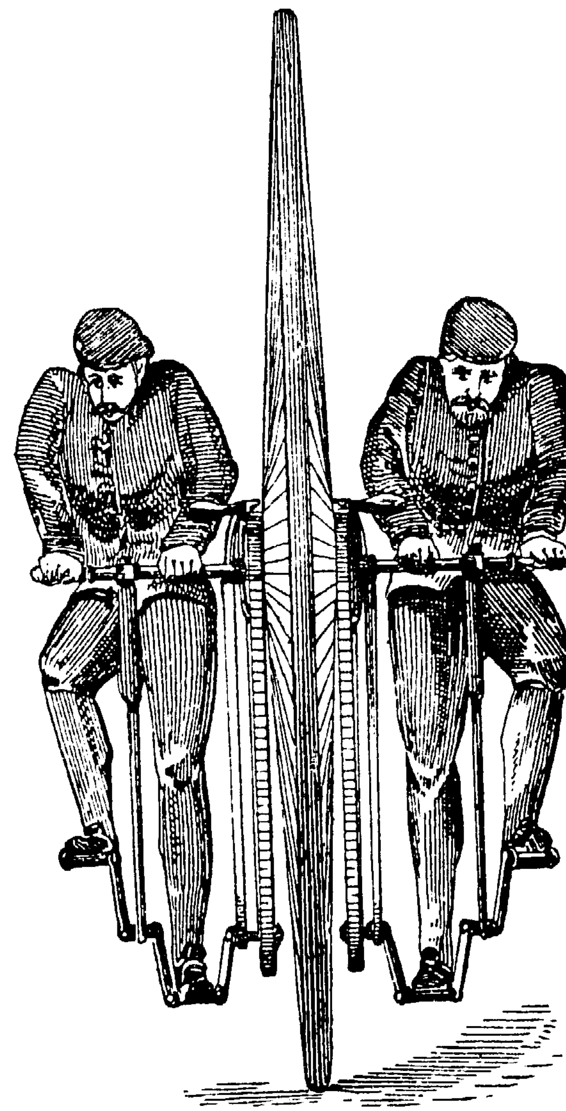


FIG. 171.

Dicycles, having two wheels mounted on a common axis. In all monocycles the transverse equilibrium is unstable; they may be subdivided into two sub-classes, according as the longitudinal equilibrium is stable or unstable. An example of the former subclass is shown in figure 169, in which the frame, carrying seat, pedal-axle, and handle, runs on an inner annular wheel, *d*, on the driving-wheel *A*; the central opening, *B*, being large enough for the body

of the rider, while his legs hang on each side of the main wheel. An example of the latter is shown in figure 170, and a sociable monocycle of the former class for two riders in figure 171.

In bicycles, the transverse equilibrium is unstable and the longitudinal equilibrium stable. In dicycles, the transverse equilibrium is stable. They may be subdivided into two sub-classes, according as their longitudinal equilibrium is stable or unstable.

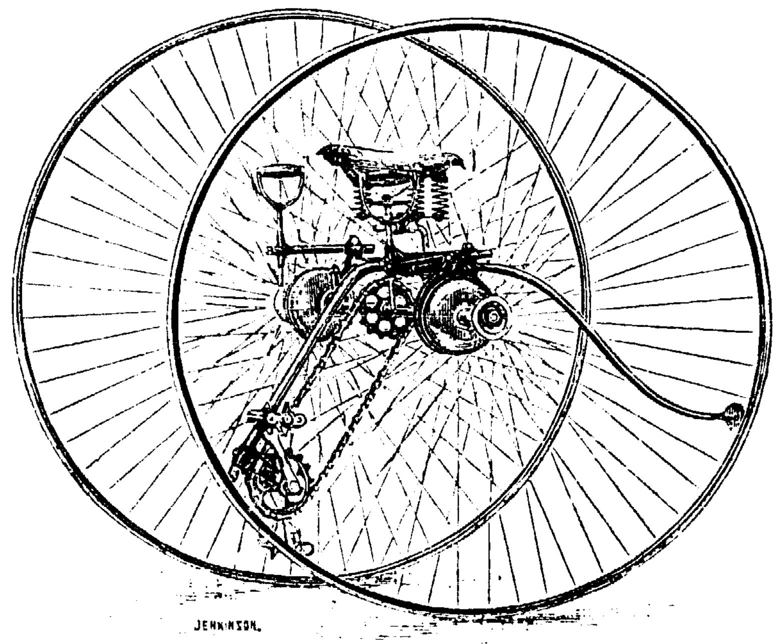


FIG. 172.

The 'Otto' dicycle (fig. 172) is the only example of the former sub-class, while none of the latter class, as already remarked, have attained any commercial importance. A dicycle of the latter type would be made with very large driving-wheels, and the mass-centre of machine and rider lower than the axis of the driving-wheel.

151. **Method of Steering.**—Proceeding to the further division and classification of bicycles, the first subdivision that suggests itself takes account of the method of steering; a bicycle being said to be a *front-* or *rear*-steerer, according as the steering-wheel is in front or behind, while among tricycles there are also *side-*steerers. A few bicycles have been made with *double-steering*.

The complete frame of the machine is usually divided into two parts, called respectively the *front-frame* and the *rear-frame*, united at the steering centre; though sometimes that part to which the saddle is fixed is called the 'frame,' to the exclusion of the other portion carrying the steering-wheel. It should be pointed out that the steering portion will sometimes be the larger and heavier of the two, the 'Humber' tricycle (fig. 149) affording an example of this. In the 'Chapman Automatic-Steering'

Safety (fig. 173) the saddle is not fixed direct to the rear-frame, but moves with the steering fork. The complete frame is in this case divided into three parts, which can move relative to each

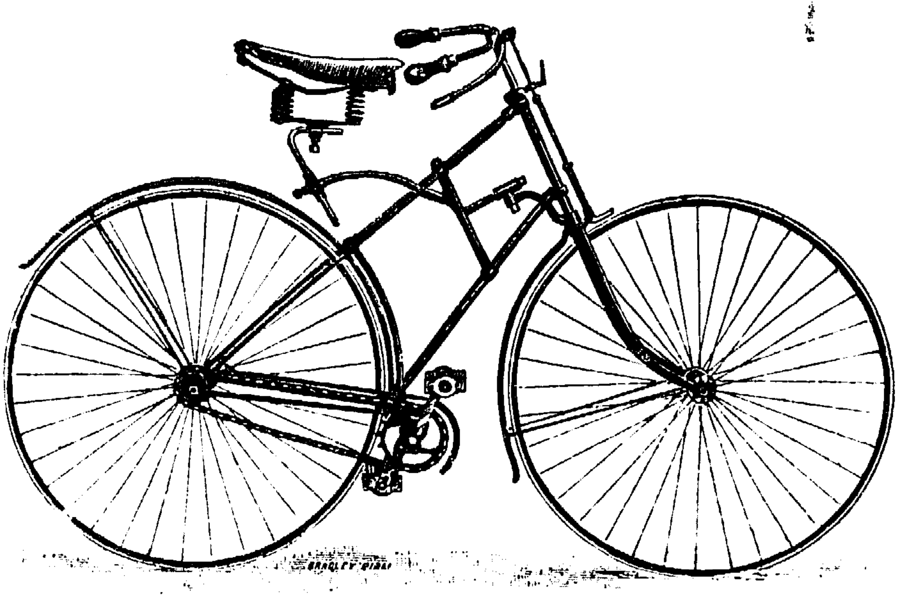


FIG. 173.

other, on which are fixed the driving-gear, the steering-wheel, and the saddle respectively.

Examples of double-steering are afforded by the 'Adjustable' Safety (fig. 174), made by Mr. J. Hawkins in 1884, and by the

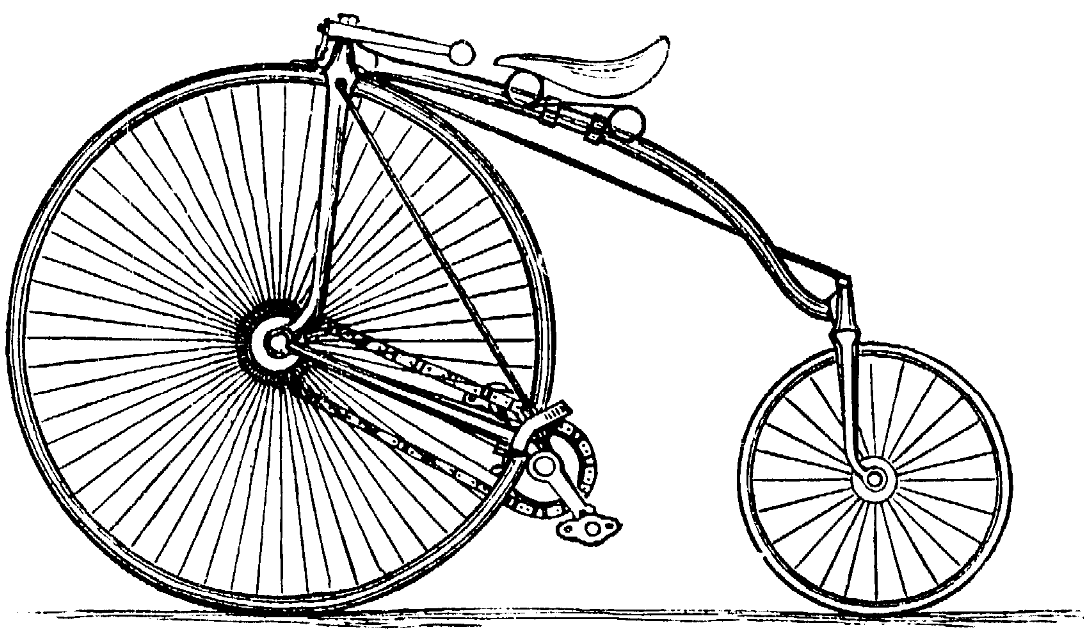


FIG. 174.

'Premier' Tandem Safety (fig. 137), in each of which the forks of both wheels move relative to the backbone.

There have been very few rear-steering bicycles made, though their only evident disadvantage is, that in turning aside to avoid an obstacle, the rear-wheel *may* foul, though the front-wheel has already cleared. Nearly all successful types of bicycles have been front-steerers.

152. **Bicycles, Front-drivers.**—Bicycles may be divided into front-drivers and rear-drivers, according to which wheel is used for driving. The 'Rucker' Tandem (fig. 135) is an example of a bicycle in which both wheels are used as drivers; but generally only one wheel is used for driving. Each of these divisions may again be subdivided into *ungeared* and *geared*.

Among ungeared front-drivers we have the 'Bone-shaker,' the 'Ordinary,' the 'Rational,' the 'Facile,' the 'Xtraordinary,' and the 'Claviger' (fig. 504). In this classification we regard as ungeared those machines in which one revolution of the driving-wheel is made for each complete cycle of the pedal's motion. Thus, any bicycle with only lever gearing will be classed as ungeared, since with such mechanism it is, in general, impossible to gear up or gear down.

Geared bicycles may be subdivided into toothed-wheel geared, chain geared, and clutch geared. Among wheel geared front-drivers we have the 'Geared Ordinary,' 'Front-Driver,' the 'Bantam,' the 'Geared Facile,' the 'Sun-and-Planet' bicycle (fig. 479), and the 'Premier' Tandem Dwarf Safety (fig. 137). Among chain geared safeties we have the 'Kangaroo,' with two driving chains, one on each side of the driving-wheel, the 'Adjustable' Safety Roadster (fig. 174), and the 'Shellard Dwarf' Safety Roadster (fig. 175).

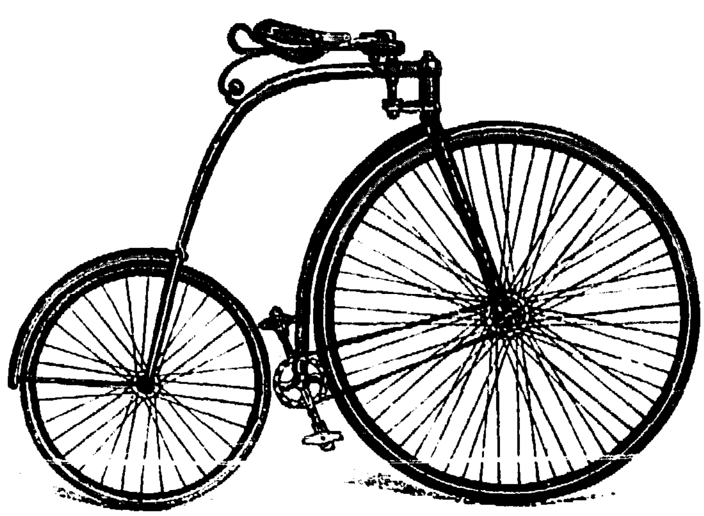


FIG. 175.

A combination of toothed-wheel and chain gear was used in the 'Marriott and Cooper' Front-Driver.

Clutch geared bicycles have never been very successful, the

Brixton Merlin Safety (fig. 176) being about the only example of this type. In the Merlin gear, a drum rotates on the axle at each side of the wheel, round which is coiled a leather strap, the other

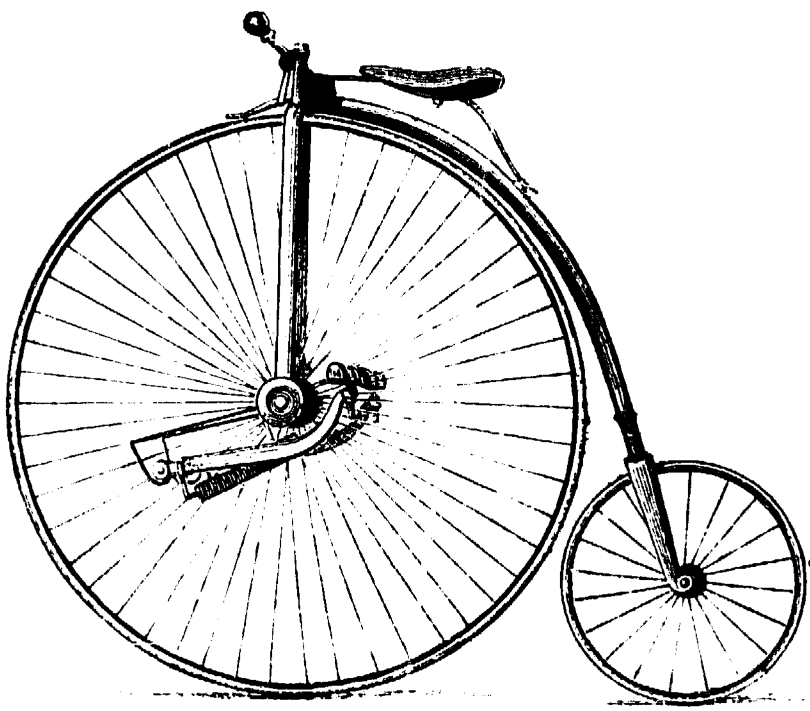


FIG. 176.

end being fastened to the pedal lever. When the pedal is pushed outwards by the rider the drum is locked by a clutch to the axle, and the effort is transmitted to the wheel. On the upstroke a spring raises the pedal lever. With this gear any length of stroke may be taken, but the imperfect action of the clutch is such that the great advantages due to the possibility of varying the length of stroke are more than neutralised.

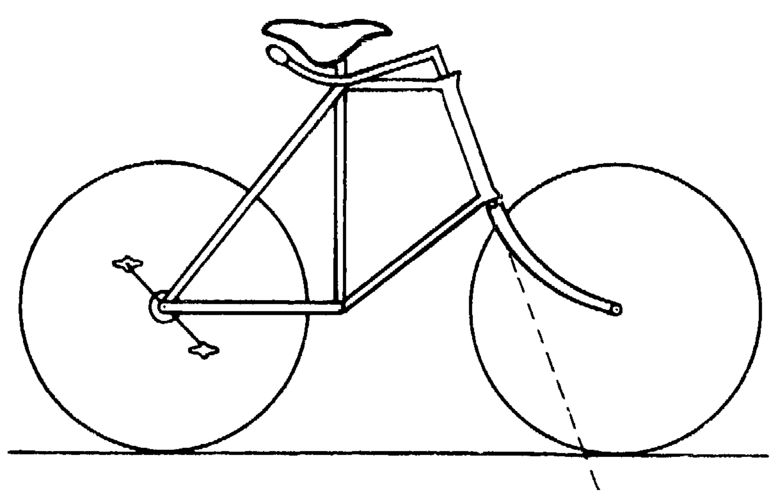


FIG. 177.

Figure 177 shows a possible front-driving rear-steering geared bicycle, the front hub having a 'Crypto' or equivalent gear.

153. **Bicycles, Rear-drivers.**—Among ungeared rear-drivers

may be mentioned the Rear-driving 'Facile' and the American 'Star' (fig. 178).

Among toothed-wheel geared rear-drivers we have the 'Burton,' the Geared 'Facile' Rear-driver (fig. 129), the 'Claviger' Geared (fig. 507), the 'Fernhead' Chainless Safety, driven by bevel-gearing. Of chain geared rear-drivers, the present popular Safety

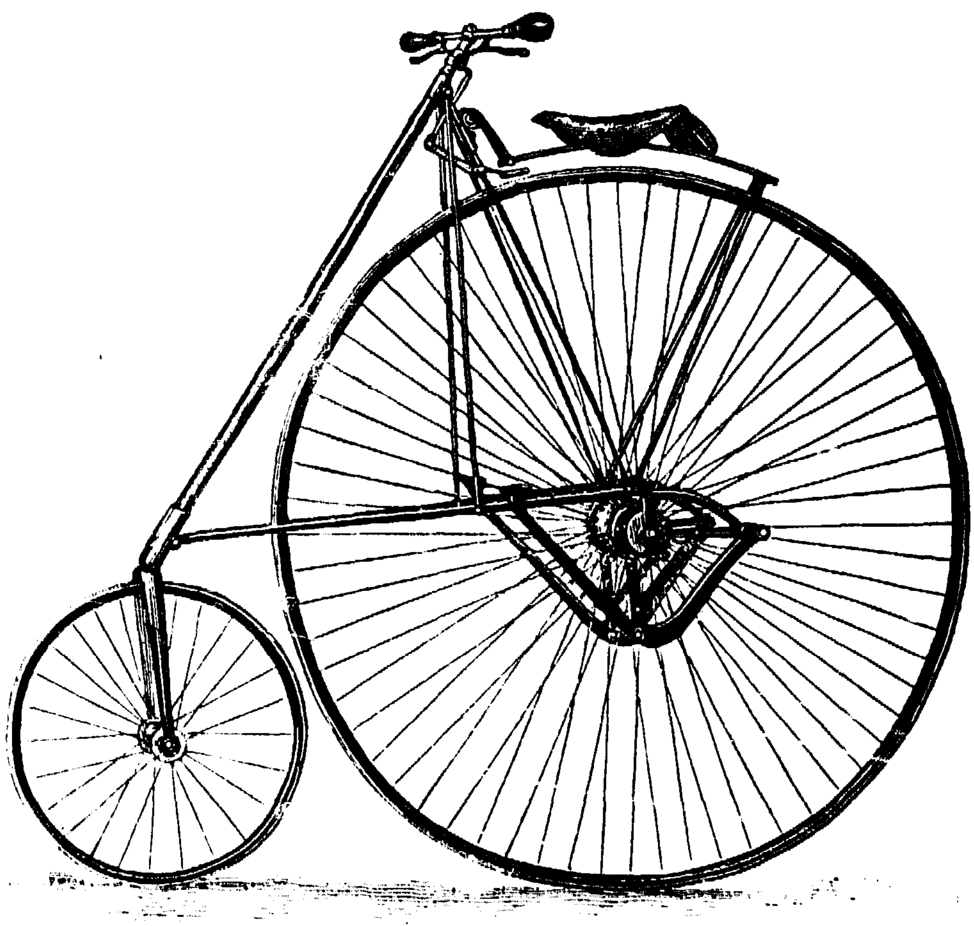


FIG. 178.

of the 'Humber' or 'Rover' type is the most important representative.

In the 'Boudard-geared' Safety a combination of toothed-wheels and chain gear is used, while the same may be said of the two-speed gears that are applied to the ordinary type of chain-driven safety.

This classification is represented diagrammatically on page 194. From this diagram it will be seen that no successful type of rear-steering bicycle has been evolved. Experimenters might with advantage direct their energies to this comparatively untrodden domain.

154. **Tricycles, Side-steering.**—The classification of tricycles may go on on similar lines to that of bicycles. There would be three types—front-steering, side-steering, and rear-steering. Of side-steering tricycles there are two subdivisions: the 'Rudge Coventry Rotary' (fig. 156) being a side-driver, while the 'Dublin' (fig. 143) and the 'Olympia' (fig. 160) are back-drivers. No side-steering, front-driving tricycle has been made, to our knowledge; though we can see nothing at present to prevent tandem tricycles of this type (figs. 179–181) from being successful roadsters. That shown in figure 179 could be ridden by a lady in ordinary costume on the front seat; it would, perhaps, be slightly deficient in lateral stability, as the mass-centre would be near the forward corner of the wheel-base triangle. That shown in figure 180 would be superior in this respect, while the weight on the driving-wheel would still probably be sufficient for all ordinary requirements. A type intermediate (fig. 181) might be made with a 'Crypto' gear on the front wheel hub, the two crank-axles being connected by a

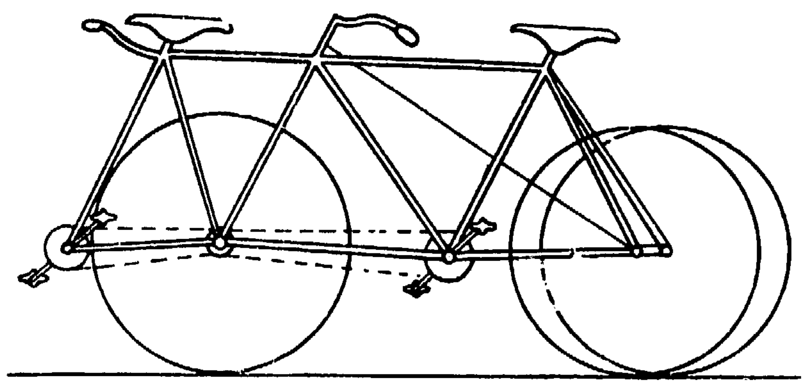


FIG. 179.

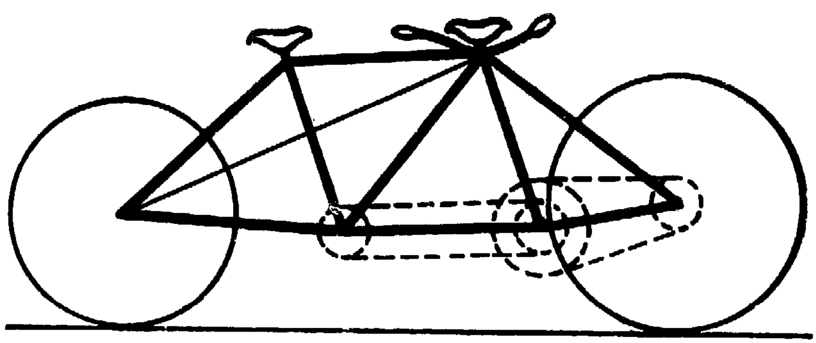


FIG. 180.

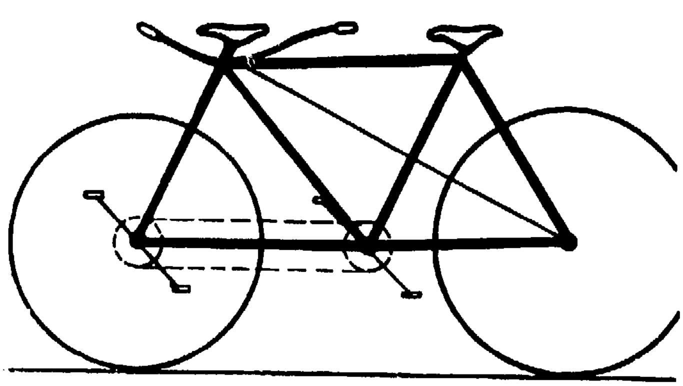


FIG. 181.

chain ; the frame would be simpler than in figures 179 and 180.

Tricycles are either single-driving or double-driving, according as there are one or two driving-wheels. The only treble-driving tricycle which has been yet put on the market is the tandem made by Messrs. Trigwell and Co., by coupling the front wheel and backbone of a 'Kangaroo' to the rear portion of a 'Cripper' (fig. 166). The driving-wheels of a double-driving tricycle are invariably mounted on the same axle, and since in going round a corner the wheels, if of equal size, must rotate at different speeds, the driving-axle must be in two parts. In the 'Cheylesmore' tricycle two separate driving chains were used between the crank- and wheel-axles, the cog-wheel on the wheel-axle being held by a clutch when driving in a straight line, while in rounding a corner the wheel which tended to go the faster overran the clutch, and all the driving effort was transmitted through the more slowly moving wheel. Starley's differential gear (see sec. 189), allowing, as it does, both wheels to be drivers under all circumstances, is now universally used for double-driving.

155. **Front-steering Front-driving Tricycles.**—The early 'Bone-shaker' tricycle (fig. 141) is an ungeared example of this class, while the 'Humber' tricycle (fig. 149) is a geared tricycle of this same class. The 'Humber' is a double-driver.

Single-driving tricycles of this division may be made by taking a 'Crypto' or 'Kangaroo' bicycle, and having two back wheels at the end of a long axle. They would, however, be deficient in lateral stability, unless used as tandems, on account of the load being applied over a point near the forward apex of the triangular wheel-base.

156. **Front-steering Rear-driving Tricycles.**—Of ungeared cycles, Lisle's early Ladies' tricycle (fig. 142) and the 'Club' (fig. 146) are examples.

The geared tricycles may be subdivided into single-drivers and double-drivers. Of the former class the 'Olympia' (fig. 160), the 'Phantom' (fig. 155), the 'Facile' (fig. 154), the 'Claviger,' and the 'Trent' convertible (fig. 182) are examples.

The double-drivers may be conveniently subdivided into direct-steerers and indirect-steerers. The 'Cripper' (figs. 150, 152, 153), of which probably more examples have been made

than all the other types put together, is a direct-steerer, so also is the Merlin (fig. 183). Among indirect-steerers we may mention the 'Devon' tricycle (fig. 145), the 'Club' (fig

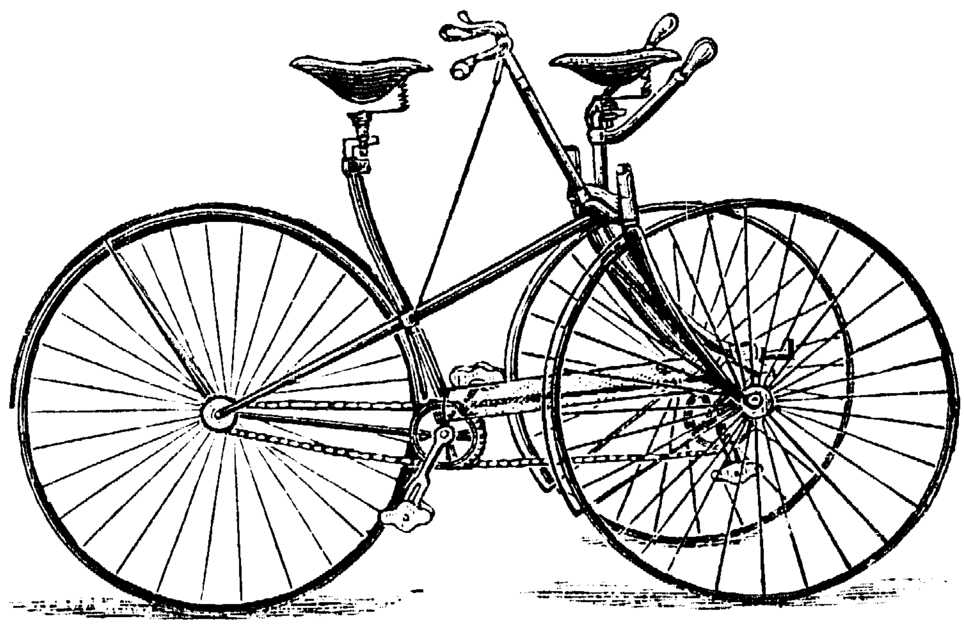


FIG. 182.

146). The 'Nottingham Sociable' (fig. 164) formed by conversion of two bicycles, and Singer's Omnicycle with clutch gear (fig. 184), made in 1879, also belong to this division.

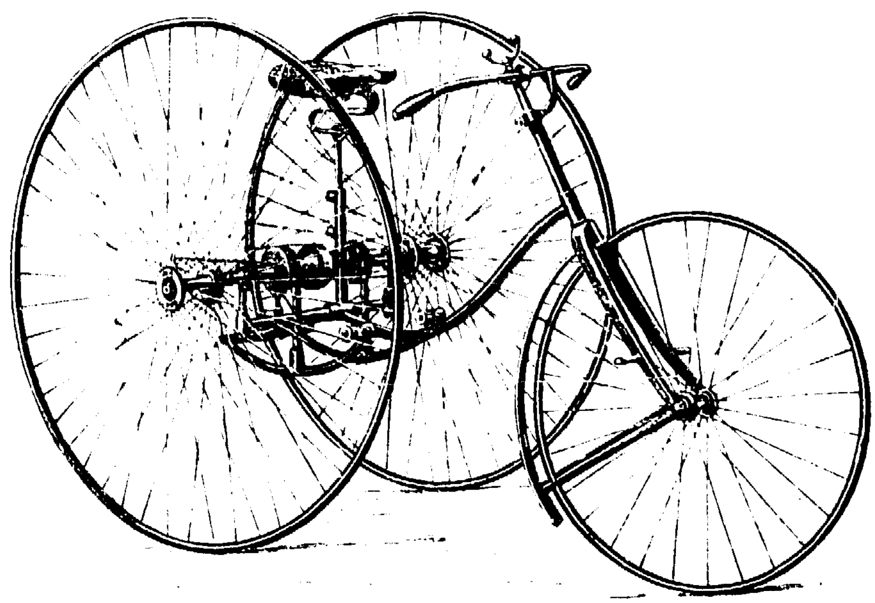


FIG. 183.

This classification of tricycles is shown diagrammatically on page 195.

157. **Rear-steering Front-driving Tricycles.**—The 'Veloci-

man,' a hand-tricycle made by Messrs. Singer & Co., of which figure 241 represents an improved design for 1896, is an example of this class. The 'Cheylesmore' (fig. 148), made by the Coventry Machinists Co., was one of the most successful of the early tricycles. Several tandem tricycles were made on this type, one of the most popular being the 'Invincible' (fig. 157), made by the Surrey Machinists Co., Limited.

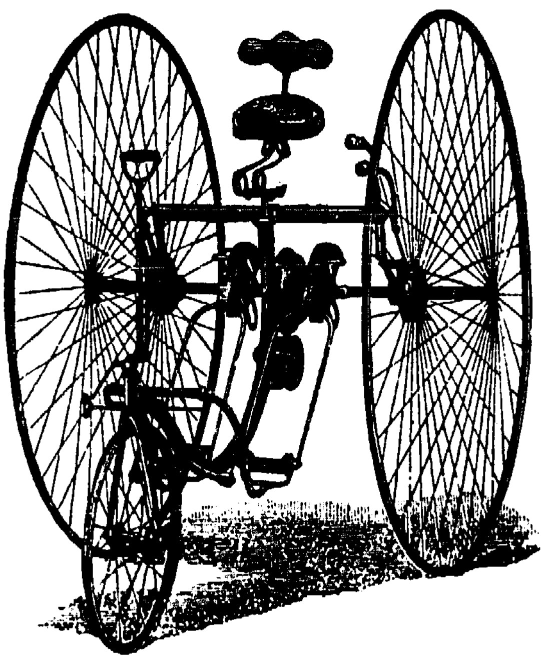


FIG. 184.

The tandem tricycles in figures 179–181, if made with both rear wheels running freely on the same axle, fixed to a rear frame, would afford examples of single-drivers of this class.

A rear-steering side-driving tricycle was the 'Challenge' (fig. 147), made by Messrs. Singer in 1879.

158. **Quadricycles.**—A great many quadricycles were made at one time by adding a piece to a tricycle, so as to form a machine for two riders (see sec. 148). The attachment of the extra portion was usually made by means of a universal joint. The one track Sociable (fig. 163) may really be classified as a four-wheel cycle, though from the lack of the universal joint in the frame it differs essentially from those mentioned above.

Rudge's quadricycle (fig. 168), giving only two tracks and a rectangular wheel base, is a very well designed machine of this type. The steering gear is similar in principle to that used in the 'Olympia' tricycle. The front portion of the frame supporting the two side-steering wheels is connected to the rear portion by a horizontal joint at right angles to the driving-axle, so that the four wheels may each touch the ground, however uneven, without straining the frame. It is made as a single, tandem, and triplet. Its stability is discussed in section 161.

The quadricycle with two tracks has some advantages as compared with the tricycle, and may well repay further consideration

CYCLES

- *Unstable equilibrium* — ONE or TWO WHEELED
  - *Transverse equilibrium unstable* — *MONOCYCLES*
    - *Longitudinal equilibrium stable* (figs. 169, 171)
    - *Longitudinal equilibrium unstable* (fig. 170)
  - *Longitudinal equilibrium stable* / *Transverse equilibrium unstable* — *BICYCLES*
  - *Transverse equilibrium stable* — *DICYCLES*
    - *Longitudinal equilibrium unstable* — Otto Dicycle (fig. 172)
    - *Longitudinal equilibrium stable* — No example
- *Stable equilibrium* — TWO WHEELED
  - *Longitudinal equilibrium stable* — No example
- *Stable equilibrium* — THREE or MORE WHEELED
  - *Statically determinate frames* — *TRICYCLES*
  - *Statically indeterminate frames* — *MULTICYCLES*

BICYCLES

- *FRONT STEERING*
  - *FRONT DRIVERS*
    - **Ungeared**
      - 'Bone-shaker' (fig. 121)
      - 'Ordinary' (fig. 122)
      - 'Rational'
      - 'Facile' (fig. 124)
      - 'Xtraordinary' (fig. 123)
      - 'Claviger' (fig. 504)
    - **Geared**
      - **Chain**
        - 'Kangaroo' (fig. 125)
        - 'Shellard' (fig. 175)
      - **Toothed Wheel**
        - 'Crypto' Geared Ordinary
        - ,, Front-Driver, 'Bantam' (fig. 132)
        - Geared 'Facile'
        - 'Sun and Planet' (fig. 479)
      - Marriott & Cooper (from Chain and Toothed Wheel)
      - **Clutch**
        - 'Merlin' (fig. 176)
  - *REAR DRIVERS*
    - **Ungeared**
      - 'American Star' (fig. 178)
      - Rear-driving 'Facile'
    - **Geared**
      - **Chain**
        - 'Humber' or 'Rover'
        - 'Quadrant' (fig. 254)
        - 'Boudard Gear' (fig. 487)
      - **Toothed Wheel**
        - 'Burton'
        - Geared 'Facile' Rear-Driver (fig. 129)
        - 'Claviger' (fig. 507)
        - 'Fernhead' Bevel-wheel
- *DOUBLE STEERING* — 'Adjustable' Safety (fig. 174)
- *REAR STEERING* — No example

TRICYCLES

- *FRONT STEERING*
  - *FRONT DRIVERS*
    - **Ungeared**
      - **Single Driving** — 'Bone-shaker' (fig. 141)
      - **Double Driving** — No example
    - **Geared**
      - **Single Driving** — No example
      - **Double Driving** — 'Humber' (fig. 149)
  - *REAR DRIVERS*
    - **Ungeared**
      - **Single Driving** — 'Facile' (fig. 154)
      - **Double Driving** — Lisle's early Ladies' Tricycle (fig. 142)
    - **Geared**
      - **Single Driving** — 'Phantom' (fig. 155); 'Claviger' Trent Convertible' (fig. 182)
      - **Double Driving**
        - **Indirect Steering** — 'Devon' (fig. 145); 'Club' (fig. 146); Two-track tricycles; 'Quadrant'; 'Nottingham Sociable' (fig. 164); Singer's 'Omnicycle' (fig. 184)
        - **Direct Steering** — 'Cripper' (fig. 150); 'Merlin' (fig. 183)
      - **Treble Driving** — (fig. 166)
- *SIDE STEERING*
  - **Side Drivers** — 'Coventry Rotary' (fig. 144)
  - **Back Drivers** — 'Dublin' (fig. 143); 'Olympia' (fig. 160)
- *REAR STEERING*
  - **Front Driving**
    - **Double Driving** — 'Invincible' Tandem (fig. 157); 'Cheylesmore' (fig. 148)
    - **Single Driving** — 'Velociman' (fig. 241)
  - **Side Driving** — 'Challenge' (fig. 147)

by cycle makers and designers. If a satisfactory mode of supporting the frame on the wheel axles by springs could be devised, the horizontal joint might be omitted, the design of frame simplified, the stability of the machine increased, and additional comfort obtained by the rider. If the two steering-wheels revolved independently on a common axle, as in the 'Phantom' tricycle (fig. 155), the design of the machine would be further simplified; the relation of the wheels to the frame being exactly the same as is a four-wheeled vehicle drawn by a horse. This type of quadricycle would, however, possess the same objectionable features as to swerving as the tricycles shown in figures 149, 154, and 155. In a horse vehicle the front axle is fixed to the shafts to which the horse is harnessed, so that the axle cannot swerve when one wheel meets an obstacle without dragging the horse sideways. In this respect the horse performs the same function as the front wheel of a 'Cripper' tricycle. A hansom cab is equivalent to a 'Cripper' tricycle, and a four-wheeler to a pentacycle (fig. 185), in which the rear portion trails after the front.

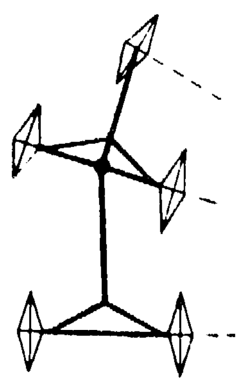


FIG. 185.

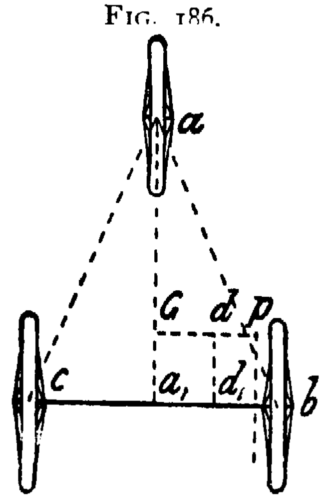
159. **Multicycles.** — By stringing together a number of 'Humber' or 'Cripper' frames with their crank-axles and pairs of driving-wheels, a cycle of 4, 6, 8, or any even number of wheels may be obtained. The steering of such a multicycle should be effected by the front rider, the intersection of the first two axles determining the radius of curvature of the path. The following wheels should be merely trailing wheels, so that they may follow in the required path.

CHAPTER XVII

STABILITY OF CYCLES

160. **Stability of Tricycles.**—If $a\,b\,c$ (fig. 186) be the points of contact of the three wheels of a tricycle with the ground, it will be in equilibrium under the action of the rider's weight, provided the perpendicular from the mass-centre of the rider and machine falls within the triangle $a\,b\,c$. If this perpendicular fall at the point d, the pressures of the wheels on the ground can easily be found by the principle of moments. Let W be the total weight of the rider and machine, w_a, w_b, and w_c the pressures of the wheels at a, b, and c on the ground. Then taking moments about the line $b\,c$, draw perpendiculars $a\,a_1$ and $d\,d_1$ to $b\,c$. We then have

FIG. 186.



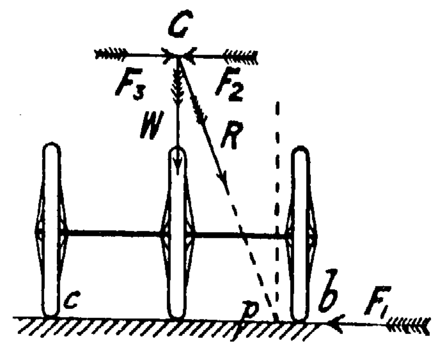


FIG. 187.

$$W \times d\,d_1 = w_a \times a\,a_1$$

or $$w_a = \frac{d\,d_1}{a\,a_1} W \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Similar expressions for w_b and w_c can be found.

If the point d fall outside the triangle $a\,b\,c$, the tricycle will topple over.

161. **Stability of Quadricycles.**—If the quadricycle be made with the steering-axle capable of turning only round a vertical axis, as in the case of an ordinary four-wheeled carriage drawn by horses, the mass-centre of the machine and rider may lie vertically

above the rectangle $a\,b\,c\,d$ (fig. 188), a, b, c and d being the points of contact of the wheels with the ground. But if one of the axles be hinged to the frame, so as to allow the four wheels to be always in contact with the ground, however uneven—as in the case of the 'Rudge' quadricycle (fig. 168)—the mass-centre of machine and rider, exclusive of front portion $a\,b$, must lie vertically above the triangle $e\,c\,d$, e being the intersection of the plans of the steering-axle and hinge joint. If the perpendicular from the mass-centre of machine and rider fall between $e\,c$ and $b\,c$, the wheel at d will lift from the ground, and the portion $e\,c\,d$ of the machine will continue to overturn until stopped by coming in contact with the portion $a\,b$.

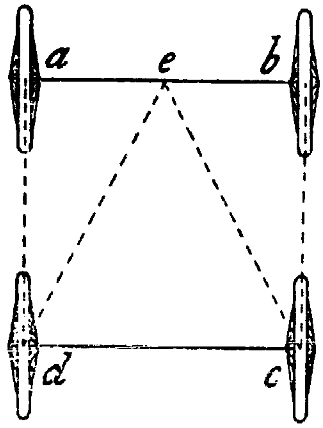


FIG. 188.

In a tandem quadricycle formed by attaching a trailing wheel, d (fig. 189), to a 'Cripper' tricycle, $a\,b\,c$, by means of a universal joint at e, the mass-centre of the machine and riders must lie vertically above and inside the quadrilateral $a\,b\,c\,d$. If the joint e be behind the axle, $b\,c$, another condition must be satisfied, viz. the vertical downward pressure at e, due to the weight on the trailing frame, must not be sufficient to tilt the triangle $a\,b\,c$ about the axle $b\,c$. This condition will in general be satisfied if the joint e be not far behind the axle.

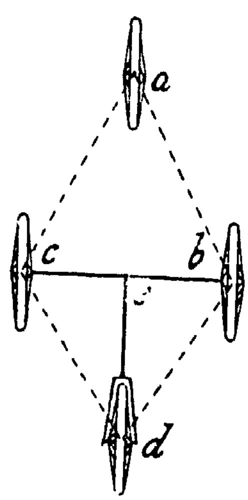


FIG. 189.

162. **Balancing on a Bicycle.**—A bicycle has only two points of contact with the ground, and a perpendicular from the mass-centre of machine and rider must fall on the straight line joining them. If the bicycle and rider be at rest, the position is thus one of unstable equilibrium, and no amount of gymnastic dexterity will enable the position to be maintained for more than a few seconds. If the mass-centre get a small displacement sideways, the displacement will get greater, and the machine and rider will fall sideways. In riding along the road with a fair speed the mass-centre is continually receiving such a displacement. If the rider

steer his bicycle in an exact straight line this displacement will get greater, and he and his bicycle will be overturned, as when at rest. But, as every learner knows, when the machine is felt to be falling to the left-hand side, the rider steers to the left—that is, he guides the bicycle in a circular arc, the centre of which is situated at the left-hand side. In popular language, the centrifugal force due to the circular motion of the machine and rider now balances the tendency of the machine to overturn ; in fact, the expert rider automatically steers the bicycle in a circle of such a diameter that the centrifugal force slightly overbalances the tendency to overturn, and the machine again regains its perpendicular position. The rider now steers for a short interval of time exactly in a straight line. But probably the perpendicular position has been slightly overshot, and the machine falls slightly to the right-hand side. The rider now unconsciously steers to the right hand, that is, in a circle having its centre to the right-hand side.

If the track of a bicycle be examined it will be found to be, not a straight line, but a long sinuous curve. With beginners the waviness of the curve will be more marked than with expert riders ; but even with the latter riding their straightest the *sinuosity* is quite apparent. A patent had actually been taken out for a lock to secure the steering-wheel of an 'Ordinary' bicycle, the purpose being to make it move automatically in a straight line. The above considerations will show, as clearly as the actual trial of his device probably did to the inventor, the absurdity of such a proceeding.

It would be possible to ride a bicycle in a perfectly straight line with the steering-wheel locked, by having a fly-wheel capable of revolving in a vertical plane at right angles to that of the bicycle wheels, and provided with a handle which could be turned by the rider. If the bicycle were falling to the right, the fly-wheel should be driven in the same direction ; the reaction on the rider and frame of bicycle would be a couple tending to neutralise that due to gravity causing the machine to fall.

Lateral Oscillation of a Bicycle.—From the above explanation of the balancing on a bicycle, it will be seen that the machine and rider are continually performing small oscillations

sideways—the axis of oscillation being the line of contact with the ground—simultaneously with the forward motion. The bicyclist and his machine may thus be roughly compared to an inverted pendulum. The time of vibration of a simple pendulum is proportional to the square root of its length, a long pendulum vibrating more slowly than a short one. In the same way, the oscillations of a high bicycle are slower than those of a low one; *i.e.* the time taken for the mass-centre to deviate a certain angle from the vertical is greater the higher the mass-centre; a rider equally expert on high and low bicycles will thus be able to keep a high bicycle nearer the exact vertical position than he will a low bicycle. In other words, the angle of swing from the vertical is greater in the 'Safety' than in the 'Ordinary.'

The track of an 'Ordinary' will therefore be straighter—that is, made up of flatter curves—than that of a 'Safety,' both bicycles being supposed ridden by equally expert riders.

163. **Balancing on the Otto Dicycle.**—In an 'Otto' dicycle at rest the mass-centre of the frame and rider is, in its normal position, vertically above the axle of the wheels; the machine is thus in stable equilibrium laterally and in unstable position longitudinally. In driving along at a uniform speed against a constant wind resistance, F (neglecting at present other resistances), the mass-centre, G, is in its normal position, a short distance, l, in front of the axle (fig. 190). While the rider exerts the driving effort the wheel exerts the force F_1 on the ground, directed backwards, and the reaction of the ground on the wheel is an equal force, F_2, in the direction of motion. The force F_2 is equivalent to an equal force F_4 at the axle and a couple Fr, r being the radius of the driving-wheel. The couple Fr is applied by the pull of the chain to the rigid body formed by the driving-wheel and axle; therefore, if T be the magnitude of this pull and r_1 the radius of the cog-wheel on the axle, $Tr_1 = Fr$.

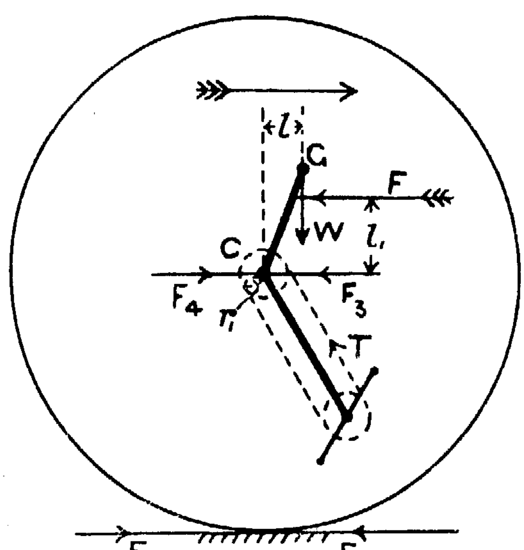


FIG. 190.

Consider now the forces acting on the rigid body formed by the frame and rider: these are, the reaction at the bearing C, the weight W acting downwards, the wind-resistance, F, and the pull of the chain T. Since the frame is in equilibrium, the moment of all these forces about any point must be zero. Taking the moments about C we get

$$W\,l - F\,l_1 = T\,r_1 = F\,r \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Suppose now the mass-centre, G, to fall a little forward of the position of equilibrium, so that the moment of W about C becomes $W\,l^1$; in order that equilibrium may be established the pull of the chain must have a greater value, T^1, thus $W\,l^1 - F\,l_1 = T^1\,r_1$. This increased pull on the chain is produced by the rider pressing harder on the pedals; in other words, by driving harder ahead.

In the same way, should the mass-centre, G, fall a little behind the position of equilibrium, the tendency to fall backward is checked by the rider easing the pressure on the pedals, *i.e.* by slightly back-pedalling.

The frame and rider in an 'Otto' dicycle thus perform oscillations about the axle of the machine; the length of the inverted pendulum is much less than in the 'Ordinary' or even the 'Safety' bicycle, and the backward or forward oscillation is greater than the lateral oscillation in a bicycle.

164. **Wheel load in Cycles when driving ahead.**—A great deal of misconception exists as to the modification of the wheel loads, due to driving ahead. If the cycle move uniformly, and the several resistances be neglected, the wheel loads will, of course, be the same as if the cycle were at rest, and therefore will depend only on the position of the mass-centre of machine and rider relative to the wheels. If the only resistance considered is the wind pressure F_1 (fig. 191), the load on the front wheel will be decreased, and that on the rear wheel increased, by the amount R, determined by the equation

$$F_1\,h_1 = R\,l\,; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

l being the wheel-base, and h_1 the distance of the centre of wind pressure above the ground. Frictional resistances, including the

friction of the bearings and gearing and the rolling friction of the wheels on the ground, make no modification of the distribution of wheel load; the former, because they are internal forces, and do not in any way affect the external forces, the latter because they act tangentially to the ground, and must be balanced by an equal and opposite reaction of the ground on the driving-wheel.

If the speed of the cycle be increased, the forces due to acceleration can be easily shown as follows: Consider the mass of the machine and rider to be concentrated at the mass-centre G, and that the wheels and frame are weightless; then, to produce the acceleration, the frame must act on the mass, and the mass react on the frame with an equal but opposite force, f. Introduce at the point of contact of the driving-wheel with the ground two equal and opposite forces, f_1 and f_2 (fig. 191), each equal and parallel to f; then f is equivalent to the force f_1, and the couple formed by the equal and opposite forces f and f_2. The force f_1 must be equilibrated by the reaction P of the ground on the driving-wheel, the couple tends to diminish the weight on the front wheel, and increases that on the rear wheel, by an amount, R, given by the equation

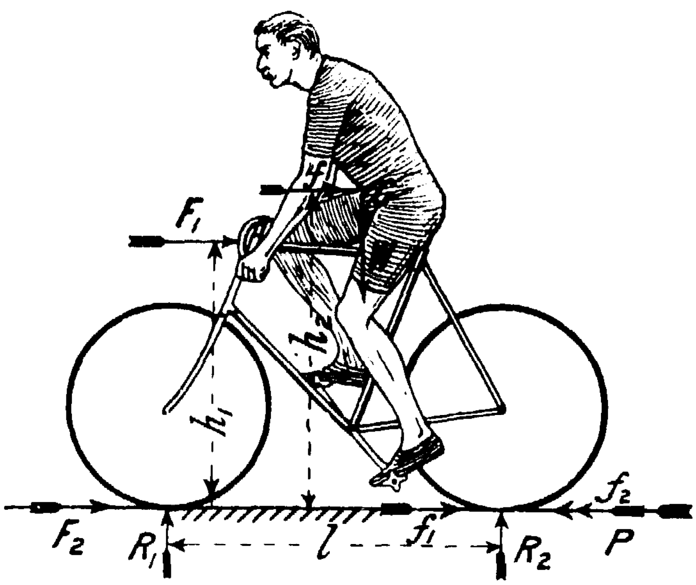


FIG. 191.

$$R\,l = f\,h_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

h_2 being the height of the mass-centre, G, above the ground.

In the most general case, the external forces acting on the system of bodies formed by the machine and rider are shown in figure 191. These are the resistance f, due to the increase of speed, the wind pressure F_1, the resistance of the wheels to rolling, F_2, the reaction of the ground on the driving-wheel, P, the weight, W, of the machine and rider, and the vertical reactions, R_1 and R_2, on the wheels. P, R_1 and R_2 are determined so as to

produce equilibrium with the other forces. Pressure exerted on the pedal does not in any way modify the reactions R_1 and R_2, except so far as it affects, or is affected by, the resistances F_1, F_2, and f; *i.e. work spent in overcoming resistances of the mechanism does not in any way affect the wheel loads.*

165. **Stability of Bicycle moving in a Circle.**—Let r be the radius of the circle in which the cycle is moving, W the weight of the rider and machine, and G the position of the mass-centre (fig. 192). We have already seen that a body of mass, W lbs., moving in a circle of radius, r, with speed v, has a radial acceleration, $\frac{v^2}{r}$; and must be acted on by a radial force $\frac{W v^2}{g r}$ lbs. Now, considering the weight of the rider and bicycle concentrated at G, and that it is transmitted from G to the ground by a weightless frame, the only forces acting on the frame are the weight W, acting vertically downwards at G, and the reaction from the ground, R. The resultant, C, of the two forces, W and R, must therefore be equal to the horizontal radial force

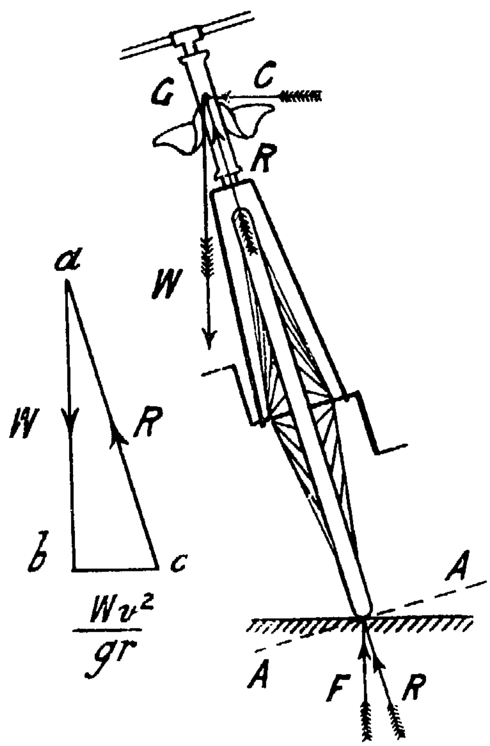


FIG. 193. FIG. 192.

$$\frac{W v^2}{g r} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

required to give the mass the circular motion, and the line of action of R must therefore pass through G. Draw $a\,b$ equal to W (fig. 193) vertically downwards, and $b\,c$ equal to $\frac{W v^2}{g r}$ horizontal. Then the reaction, R, is represented in magnitude and direction by $c\,a$. When the rider is moving steadily in a circle the machine must be inclined at the angle $c\,a\,b$ to the vertical, so that the reaction, R, from the ground may pass through G (see sec. 45).

166. **Friction between Wheel and Ground.**—When there

is no friction between two surfaces in contact the mutual pressure is at right angles to the surfaces. Any component of force parallel to the common surface of contact can only be due to friction. In the case of a bicycle moving in a circle, the centripetal force is supplied by the friction between the wheel and the ground. If the surface of the road be greasy, the friction is insufficient to provide the proper amount of force, and the force of reaction of the ground, F, together with the weight of the machine and rider, W, form a couple (fig. 192) tending to overturn the machine.

Now when a couple acts on a rigid body free to move, the body turns about its mass-centre (see sec. 66). In the case of the bicycle (fig. 192), the mass-centre, G, will have a simultaneous motion downwards, so that the final result will be that the wheel will slip to the right.

Figure 192 also illustrates the forces acting on a bicycle which is being steered in a straight line, and which has already attained a slight inclination to the vertical; the weight, W, of the rider and the reaction of the ground, F, form a couple tending to increase still further the deviation from the vertical.

167. **Banking of Racing Tracks.**—In racing tracks, the surface of the ground at the corners is sloped, as at A A (fig. 192), so as to be perpendicular to the average slope of the bicycles going round the corner. From (5) it is evident that this slope depends on the speed of the cyclists and the radius of the track. Table VIII. gives the necessary slopes for different speeds and radii of track.

Example.—Taking a speed of twenty-four miles per hour and the radius of the track 160 feet, $v = \frac{24 \times 5280}{3600} = 35{\cdot}2$ ft. per second, $\frac{W v^2}{g r}$ becomes $\frac{35{\cdot}2^2}{32{\cdot}2 \times 160} W = {\cdot}24\ W$; that is, $b\,c = {\cdot}24\ a\,b$ (fig. 193), and therefore the surface of the track must be laid at a slope of 24 vertical to 100 horizontal. If the track be laid at this slope, the wheel of a bicycle moving at a less speed than twenty-four miles an hour will tend to slip downwards towards the inside of the track, that of a bicycle moving at a higher speed will tend to move upwards towards the outside.

TABLE VIII.—BANKING OF RACING TRACKS.

Parts Vertical Rise in 100 *Parts Horizontal.*

Mean radius of track	Speed, miles per hour. 20	25	30	35	40
50 ft.	53·4	83·4	120·2	163·7	213·7
100 ft.	26·7	41·7	60·1	81·7	106·8
150 ft.	17·8	27·8	40·1	54·5	71·2
200 ft.	13·3	20·9	30·0	40·9	53·4
250 ft.	10·7	16·7	24·0	32·7	42·7
300 ft.	8·9	13·9	20·0	27·2	35·6

If the width of the track be considerable, the slope should be greater at the inner than at the outer edge, for a given speed. In

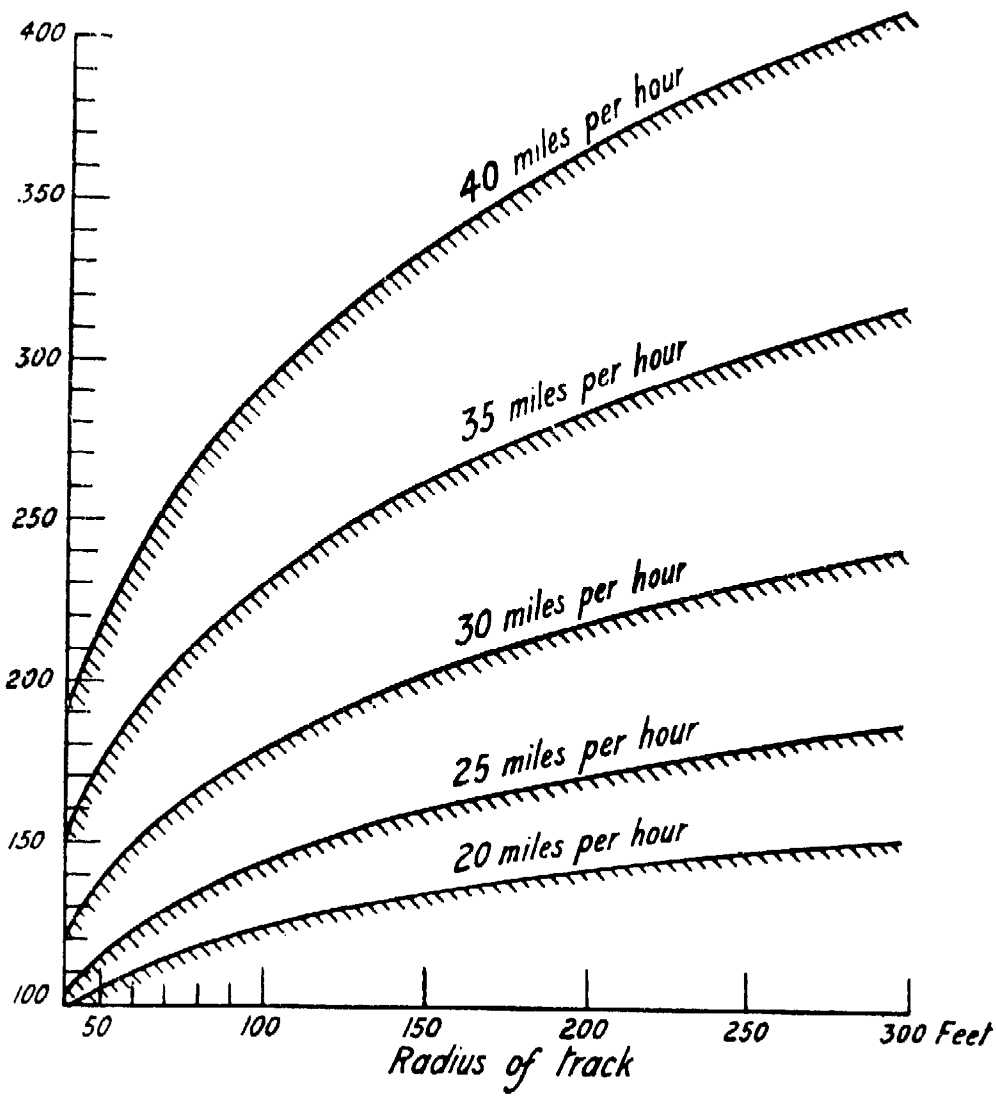


FIG. 194.

this case it can be shown by an easy application of the integral calculus, that if R be the radius at any point of the track and

y the corresponding height above a certain horizontal datum level

$$y = \frac{v^2}{g} \log_e R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

feet and seconds being the units.

If V be the speed in miles per hour,

$$y = \cdot 15383 \ V^2 \log R^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

y and R being in feet, and log R being the ordinary tabular logarithm.

Table IX. contains the values of y for different values of R from 40 to 300 feet, and at various speeds from 20 to 40 miles per hour, and figure 194 shows cross sections of tracks for these various speeds.

TABLE IX.--BANKING OF RACING TRACKS.

Elevation above a datum level, in feet.

Radius feet	Speed, miles per hour				
	20	25	30	35	40
40	98·4	153·8	221·5	301·4	393·7
50	104·5	163·3	235·2	320·1	418·1
60	109·4	170·9	246·2	335·0	437·6
70	113·5	177·4	255·4	347·6	454·1
80	117·1	183·0	263·5	358·6	468·4
90	120·2	187·9	270·5	368·2	481·0
100	123·1	192·3	276·9	376·9	492·2
110	125·6	196·3	282·6	384·6	502·4
120	127·9	199·9	287·8	391·8	511·7
130	130·5	203·2	292·6	398·3	520·2
140	132·0	206·3	297·1	404·3	528·1
150	133·9	209·2	301·3	410·0	535·6
175	138·0	215·6	310·5	422·6	552·0
200	141·6	221·2	318·6	433·6	566·3
225	144·7	226·1	325·6	443·2	578·9
250	147·5	230·5	332·0	451·8	590·1
275	150·1	234·5	337·7	459·4	600·3
300	152·4	238·1	342·9	466·7	609·6

Since the circumference of the inner edge of the track is less than that of the outer edge, when record-breaking is attempted, the rider keeps as close as he safely can to the inner edge ; conse-

quently the average speed of riding is greatest at the inner edge. On this account, the convexity of the cross-section is, with advantage, made greater than shown in figure 194.

168. **Gyroscopic Action.**—In the above investigation, it has been assumed that the weight of the wheels is included in that of the rider and machine, and no account has been taken of their gyroscopic action. We have already seen (sec. 70) that if a wheel, of moment of inertia I, have a rotation, ω, about a horizontal axis, and a couple, C, be applied to the axle tending to make it turn in a vertical plane, the axle will actually turn in a horizontal plane with an angular velocity of precession

$$\theta = \frac{C}{I\omega} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Thus, in estimating the stability of a wheel rolling along a circular arc, both centrifugal and gyroscopic actions must be considered.

Let R be the radius of the track described by the bicycle, r the outside radius and r_1 the radius of gyration of the wheels, V the speed of the cyclist, and w the weight of the wheels; then

$$\theta = \frac{V}{R}, \ I = \omega r_1^2, \ \omega = \frac{V}{r}.$$

Substituting in formula (8) we get

$$C = \frac{w V^2 r_1^2}{R r} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

i.e. the gyroscopic couple required, in addition to the centrifugal couple, is proportional to the square of the speed, inversely proportional to the radius of the track, and approximately proportional to the radius of the cycle wheels.

Example.—If the total weight of the machine and rider be 180 lbs., the weight of the wheels 8 lbs., speed 30 miles per hour =44 feet per sec., the radius of the track 100 feet, r the radius of the wheel 14 in. = $\frac{14}{12}$ feet, and r_1 = 13 in. = $\frac{13}{12}$ feet, we get V = 44 ft. per sec., and

$$C = \frac{8 \times 44^2 \times 13^2}{100 \times 14 \times 12} = 155{\cdot}8 \text{ foot-poundals}$$

$$= 4{\cdot}84 \text{ foot-lbs.}$$

i.e. the mass-centre of the machine and rider will have to be $\frac{4 \cdot 84}{180} = \cdot 027$ feet, or ·32 inches further from the vertical than if the wheels were weightless, and gyroscopic action could be neglected.

From the above example it will be seen that gyroscopic action in bicycles of the usual types is negligible, except at the highest speeds attainable on the racing-path, and on tracks of small radius. If a fly-wheel were mounted on a bicycle and geared higher than the driving-wheel, the gyroscopic action might be, of course, increased. If the fly-wheel were parallel to, and revolved in the same direction as the driving-wheel, the rider, while moving in a circle, would have to lean further over than would be necessary without the fly-wheel. If, on the other hand, the fly-wheel revolved in the opposite direction, the rider would have to lean over a less distance; in fact, by having the $I\omega$ of the fly-wheel large enough it might be possible for a bicyclist to keep his balance while leaning towards the *outside* of the curve being described.

The same gyroscopic action takes place when a tricycle moves in a circle.

169. **Stability of a Tricycle moving in a Circle.**—A tricycle moving round a curve is subjected to the same laws of centrifugal force as a bicycle, the only difference being that the frame of the machine cannot tilt so as to adjust itself into equilibrium with the forces acting.

Let figure 186 be the plan and figure 187 the elevation of a tricycle moving in a circle, the centre of which lies to the left. Let G be the mass-centre of the machine and rider, a, b and c the points of contact of the wheels with the ground. Considering the mass of the machine and rider concentrated at G, a horizontal force, F_2, applied at G is necessary to give the body its circular motion. This force is supplied by the horizontal component of the reaction of the saddle on the rider. There will be an equal horizontal force, F_3, exerted on the frame at G_1 by the rider. This force tends to make the wheels slip sideways on the ground, an equal but oppositely directed force, F_1, will be exerted by the ground on the wheels. The force F_2 gives the body its necessary radial acceleration, while the forces F_1 and F_3 acting on the machine form a couple tending to overturn it. If the resultant R

of the forces F_3 and W cut the ground at a point, p, outside the wheel base, $a\,b\,c$, the machine will overturn. Hence the necessity for tricyclists leaning over towards the inside of a curve when moving round it.

Again, if the force F_1 be greater than $\mu\, W$, the tricycle will slip bodily sideways, μ being the co-efficient of sliding friction between the tyres and the ground. This slipping is often experienced on greasy asphalte or wood paving.

170. **Side-slipping.**—The side-slipping of a bicycle depends on the coefficient of friction between the wheels and the ground, and the angle of inclination of the bicycle to the vertical. The coefficient of friction varies with the condition of the road, being very low when the roads are greasy ; when the roads are in this condition the bicyclist, therefore, must ride carefully. The condition of the roads is a matter beyond his control, but the other factor entering into side-slipping is quite within his control. In order to avoid the chance of side-slipping, no sharp turns should be made on greasy roads at high or even moderate speeds. To make such turns, we have seen (sec. 165) that the bicycle must be inclined to the vertical, this slope or inclination increasing with the square of the speed and with the curvature of the path. At even moderate speeds this inclination is so great that on greasy roads there would be every prospect of side-slipping taking place. If a turn of small radius must actually be effected, the speed of the machine must be reduced to a walking pace or even less.

A well-made road is higher at the middle than at the sides. When riding straight near the gutter the angle made by the plane of the bicycle with the normal to the surface of the ground is considerable. If the rider should want to steer his bicycle up into the middle of the road, in heeling over this angle is increased. This may be safely done when the road is dry, but on a wood pavement saturated with water it is quite a dangerous operation. With the road in such a condition the cyclist should ride, if traffic permit, along its crest.

The explanation given above (sec. 162) that in usual riding the lateral swing of a 'Safety' is greater than that of an 'Ordinary,' explains why side-slipping is more often met with in the lower machines. The statement of some makers that their particular

arrangement of frame, gear, or tread of pedals, &c., prevents side-slipping is utterly absurd; the only part of the machine which can have any influence on the matter being the part in contact with the ground—that is, the tyres. Again, the statement of riders that their machines have side-slipped when going straight and steadily cannot be substantiated. A rider may be going along quite carefully, yet if his attention be distracted for a moment, and he give an unconscious pull at the handles, his machine may slip.

Side-slipping with Pneumatic Tyres.—A pneumatic tyre has a much larger surface of contact with the ground than the old solid tyre of much smaller thickness. This fact, which is in its favour as regards ease of riding over soft roads, is a disadvantage as regards side-slipping on greasy surfaces. The narrow tyre on a soft road sinks into it, the bicycle literally ploughing its way along the ground; and on hard roads the narrow tyre is at least able to force the semi-liquid mud from beneath it sideways, until it gets actual contact with the ground. The pressure per square inch on the larger surface of a pneumatic tyre in contact with the ground being very much smaller, the tyre is unable to force the mud from beneath it; it has no *actual* contact with the ground, but floats on a very thin layer of mud, just as a well lubricated cylindrical shaft journal does not actually touch the bearing on which it nominally rests, but floats on a thin film of oil between it and the bearing. The coefficient of friction in such a case very small, and a slight deviation of the bicycle from the vertic position—*i.e.* steering in any but a very flat curve—may cau side-slip.

The non-slipping covers, now almost entirely used on roadst pneumatic tyres, are made by providing projections of such sm area that the weight of the machine and rider presses the through the thin layer of mud into actual contact with the groun The coefficient of friction under these circumstances is high , and the risk of side-slip correspondingly reduced.

Apparent Reduction of Coefficient of Friction.—While the driving-wheel rests on a greasy road a comparatively small driving force may cause the wheel to slip circumferentially on the road, instead of rolling on it. This skidding of the wheel, though

primarily making no difference in the conditions of stability, in a secondary manner influences side-slipping considerably.

Let a body *M* (fig. 195) of weight *W*, resting on a horizontal plane, be acted on by two horizontal forces, *a* and *b*, at right angles. Let μ be the coefficient of friction, and let at first only one of the forces, *b*, be in action. To produce motion in the direction *M X*, *b* must be greater than μW. Now, suppose the body *M* is being driven, under the action of a force *a*, in the direction *M Y*, in this case a much smaller force, *b*, will suffice to give the body a component motion in the direction *M X*. The actual motion will be in the direction *M R*, and since friction always acts in a direction exactly opposite to that of the motion, the resultant force on the body *M* must be in the direction *M R*. Let *F* be this resultant force; its components in the directions *M X* and *M Y* must be *b* and *a* respectively. Now, if the force *a* be just greater than μW, it will be sufficient to cause the body to move in the direction *M Y*, and any force, *b*, however small, will give *M* a component motion in the direction *M X*.

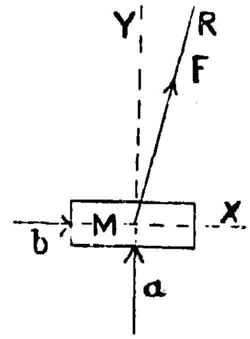


FIG. 195.

A familiar example illustrating the above principle, which has probably been often put into practice by every cyclist, is the adjusting of the handle-pillar in the steering-head. If the handle-pillar fits fairly tightly, as it ought to do, a direct pressure or pull parallel to its axis may be insufficient to produce the required motion, but if it be twisted to and fro—as can easily be done on account of the great leverage given by the handles—while a slight upward or downward pressure is exerted, the required motion is very easily obtained.

In the 'Kangaroo' bicycle the weight on the driving-wheel was less than in either the 'Rover Safety' or in the 'Ordinary.' On greasy roads it was easy to make the driving-wheel skid circumferentially by the exercise of a considerable driving pressure. This circumferential slipping once being established, the very smallest inclination to the vertical would be sufficient to give the wheel a sideway slip, which would, of course, rapidly increase with the vertical inclination of the machine.

171. **Influence of Speed on Side-slipping.**—The above dis-

cussion on side-slipping presumes that the speed of the machine and rider is not very great, so that the momentum of moving parts does not seriously influence the question. If the speed be very great, however, the momentum of the reciprocating parts, due principally to the weight of the rider's legs, pedals, and part of the weight of the crank, may have a decided influence on side slipping.

Let G be the mass-centre of the machine and rider (fig. 196), let the total mass be W lbs., let the linear speed of the pedals relative to the frame of the machine be v, and let w be the mass in lbs. of one of the two bodies to which the vertical components of the pedals' velocity is communicated : w will approximately be made up of the pedal, half the crank, the rider's shoe, foot, and leg from the knee downwards, and about one-third of the leg from the knee to the hip-joint. If the rider's ankle-action be perfect, the mass w may be considerably less, depending on the actual vertical speeds communicated to the various portions of the leg. Let the centre of the mass w be distant l_1 from the central plane of the bicycle. When the pedal is at the top of its path this mass possesses no velocity in a vertical direction, and therefore no vertical momentum. When the crank is horizontal and going downward, the vertical velocity is at its maximum, and the momentum is $w\,v$. Let t be the time in seconds taken to perform one revolution of the crank, the time taken to impress this momentum is $\frac{t}{4}$; and if f^1 be the *average* force in poundals acting during this time to produce the change, we must have (sec. 63) :

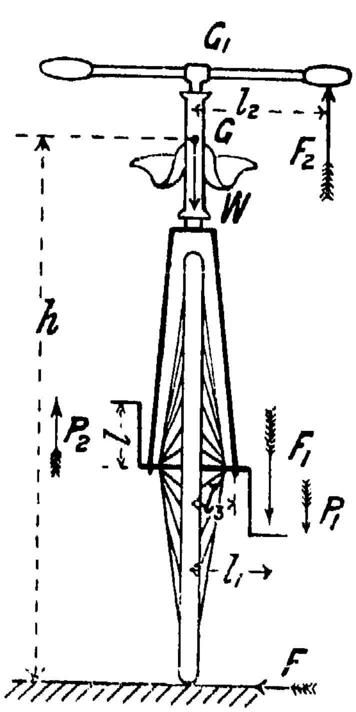


FIG. 196.

$$f^1 \frac{t}{4} = w v.$$

Therefore $$f^1 = \frac{4\,w\,v}{t}.$$

If f be the average force in lbs., $f^1 = gf$, and the above equation may be written,

$$f = \frac{4wv}{gt} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

If l be the length of the crank, the length of the path described in one revolution by the pedal-pin is $2\pi l$, and the time taken to perform one revolution is $\frac{2\pi l}{v}$. Substituting in (10) we get,

$$f = \frac{2wv^2}{g\pi l} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Now leaving out of consideration for an instant the action of any force at the point of contact of the machine with the ground, and considering the machine and rider as forming one system, the above force f is an internal force, and can thus have no action on the mass-centre, G, of the whole system. But two parts of the system have each been impressed with a moment of momentum, $w v l_1$, about the mass-centre G, the remaining part $(W - 2w)$ will be impressed with a momentum numerically equal but of opposite sense. Let G_1 be the mass-centre of this remaining part. Then the up-and-down motion of the two pedals being as indicated by the arrows p_1 and p_2, the point G_1 must move to the left with a velocity, v_1, such that

$$2wvl_1 = (W - 2w)v_1 \times \overline{GG_1}.$$

Thus, if there be absolutely no friction between the wheel and the ground, the point of contact of the wheel must slip sideways to the right.

Let F be the *average* frictional resistance, in lbs., required to prevent this slipping, then

$$Fh = 2fl_1,$$

or

$$F = \frac{4wv^2 l_1}{g\pi l h} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

If n be the number of turns per second made by the crank, $v = 2\pi n l$, and (12) may be written

$$F = \frac{16\pi n^2 l_1 l w}{gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

From (12) and (13) the lateral force F, or what may be called the 'tendency' to side-slip, is proportional to the masses which partake of the vertical motion of the pedals, to the width of the tread, and inversely proportional to the height of the mass-centre from the ground; from (12) it is proportional to the square of the speed of the pedals, and inversely proportional to the length of the crank; from (13) it is proportional to the square of the number of revolutions of the crank-axle and to the length of the crank.

The force F changes in direction twice during one revolution of the crank-axle. It is equivalent to an equal force acting at G, and a couple Fh. The force acting at G, changing in direction, will therefore cause the mass-centre of the bicycle and rider to move in a sinuous path, even though the track of the wheel be a perfectly straight line. The less this sinuosity, other things being equal, the better; *i.e.* in this respect a high bicycle is better than a low one for very high speeds.

It must be carefully noted that in the above investigation the pressure exerted on the pedal by the rider does not come into consideration. When moving at a given speed the tendency to side-slip is therefore quite independent of whether pressure is being exerted on the pedal or not.

172. **Pedal Effort and Side-slip.**—The idea that the pressure on the pedal causes a tendency to side-slip is so general that it may be worth while to study in detail the forces acting on the rider, the wheel and pedals, and the frame of the machine. For simplicity we will consider an 'Ordinary,' in which the rider is vertically over the crank-axle. The investigation will be of the same nature, but a little longer, for a rear-driving 'Safety.' The weight of the machine will be neglected.

Let W be the weight of the rider, F_1 the vertical thrust on the pedal, F_2 the upward pull on the handle-bar, F_3 the vertical pressure on the saddle; let l_1 and l_2 be the distances of the lines of action of F_1 and F_2 respectively, and l_3 the distance of the crank axle-bearing from the central plane of the machine (fig. 196).

Consider first the forces acting on the rider; these are, his weight, W, acting downwards at G; the pull, F_2, of the handle-

bar downwards; the reaction, F_1, from the pedal upwards; and the reaction, F_3, of the saddle. These forces are all parallel, and since the rider is in equilibrium we must have

$$W - F_1 + F_2 - F_3 = 0 \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Also, the moments of these forces about any point is zero; therefore, taking moments about the mass-centre, G, if the rider has not shifted sideways when exerting the pressure F_1 on the pedals,

$$F_1 l_1 - F_2 l_2 = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

If the rider does not pull at the handles he must either grip tightly on to the saddle, or shift sideways, so that the moment of the force F_1 is balanced.

Consider next the forces acting on the frame, which, for clearness of illustration, is shown isolated (fig. 197); these are, the pull, F_2, on the handle-bar upwards; the pressure, F_3, of the rider on his saddle downwards; and the upward reaction of the bearings f_1 and f_2. These forces are all parallel, and since they are in equilibrium,

$$F_2 - F_3 + f_1 + f_2 = 0\,;$$

that is,

$$f_1 + f_2 = F_3 - F_2 \quad . \quad . \quad (16)$$

Since the force $(F_3 - F_2)$ has no horizontal component, neither will the force $(f_1 + f_2)$. By taking moments of all the forces about the point of application of f_2, the value of f_1 may be found, and then f_2 can be determined.

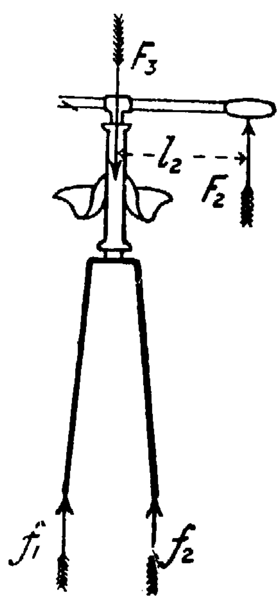


FIG. 197.

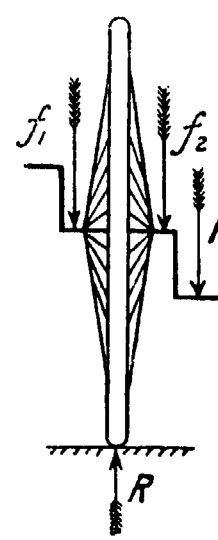


FIG. 198.

Now, consider the forces acting on the wheel (fig. 198), including cranks and pedal-pin, which together form one rigid body. Besides the forces F_1, f_1, and f_2, there is only the reaction of the ground, R, and since the wheel is in equilibrium vertically,

$$R - F_1 - f_1 - f_2 = 0.$$

Substituting the value of $f_1 + f_2$ from (16) we get

$$R = F_1 - F_2 + F_3 = W \quad . \quad . \quad . \quad . \quad (17)$$

R being vertical, there is no tendency to side-slip.

The above result can be more simply obtained, thus: considering the bicycle and rider as forming one system of bodies, the external forces acting are in equilibrium; and since these consist only of the weight, W, and the reaction, R, R must be (sec. 71) equal, parallel but opposite to W. W being vertical, R must also be vertical. The force F_1 exerted by the rider on the pedal is an internal force, and has not the slightest influence on the external forces acting on the system.

173. **Headers.**—Taking a 'header' over the handle-bar was quite an every-day occurrence with riders of the 'Ordinary' bicycle. In the 'Ordinary,' the mass-centre of the rider and machine was situated a very short distance behind a vertical through the centre of the front wheel, so that the margin of stability in a forward direction was very small; any sudden check to the progress of the machine by an obstruction on the road, by the rider applying the brake, or back-pedalling, was in many cases sufficient to send him over the handle-bar. Two classes of headers have to be distinguished: (I) That in which the front wheel may be considered rigidly fixed to the frame; the header being caused either by the application of the brake to the front wheel, or by back-pedalling in a Front-driver. (II) That in which the front wheel is quite free to revolve in its bearings; the header being caused by an obstruction on the road, application of the brake to the back wheel, or back-pedalling in a Rear-driver.

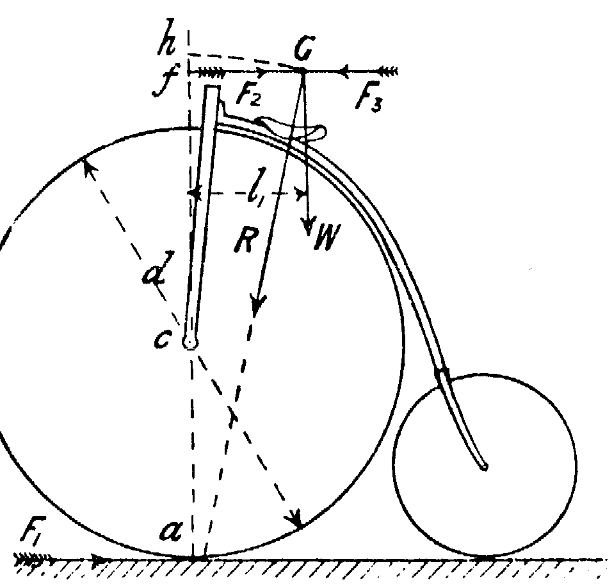


FIG. 199.

(I) Let l_1 (fig. 199) be the distance of the mass-centre, G, from a vertical through the wheel centre, c; then, in order that the wheel, frame, and rider may turn as one body about

the point a as centre, a moment, $W l_1$, must be applied. If d be the diameter of the driving-wheel, μ the coefficient of friction of the brake, and P the pressure of the brake just necessary to lock the frame on the wheel and so cause a header,

$$\frac{\mu P a}{2} = W l_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

If the pressure actually applied to the brake be equal to or greater than P, determined by the above equation, the wheel will be locked to the frame.

Let the circle through G with centre a cut the vertical through c at h. From G draw a horizontal to cut $c h$ in f. In taking a header, the weight of the machine and rider has to be lifted a distance $f h$. If v be the speed of the machine, the kinetic energy stored up in it is $\frac{W v^2}{2 g}$, and the work done in lifting it through the height $f h$ is $W \times f h$; therefore, if the speed v be greater than that determined by the formula

$$\frac{v^2}{2 g} = f h \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

a header will occur if the brake-pressure be applied strongly.

If the check to the speed of a Front-driver be made by back-pedalling, r be the radius of the crank, and P_1 the back-pedalling force applied, we have,

$$P_1 r = W l_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

The action of back-pedalling in a Front-driver is the same as that of applying the brake to the front wheel, as regards the locking of the front wheel to the frame. The speed at which a header will occur if vigorous back-pedalling be applied is in this case also given by equation (19).

Example I.—In a 54-inch 'Ordinary,' the point G (fig. 199) may be 60 inches above the ground and 10 inches behind the wheel-centre c. The height, $f h$, will then be about 1·2 inch $= \frac{1}{10}$ foot. Substituting in (19)

$$\frac{v^2}{2 \times 32{\cdot}2} = \frac{1}{10}, \text{ from which } v = 2{\cdot}5 \text{ feet per second,}$$

$$= 1{\cdot}9 \text{ mile per hour.}$$

Example II.—In a 'Safety' (fig. 200) the height, fh, may be 2 feet. Substituting in (19),

$$\frac{v^2}{2 \times 32 \cdot 2} = 2, \text{ from which } v = 11 \cdot 1 \text{ feet per second,}$$
$$= 7 \cdot 6 \text{ miles per hour.}$$

The subject may be looked at from another point of view. Let F_1 be the horizontal force of retardation which must be supplied by the action of the ground on the wheel. This is transmitted through the wheel, so that an equal force, F_2, acts on the mass at G, and the mass reacts on the frame with an equal and opposite force, F_3. Then, in order that stability may be maintained, the resultant R of W and F_3 must not cut the ground in advance of the point of contact a. If R cuts the ground in front of a, the machine will evidently roll over about a as centre.

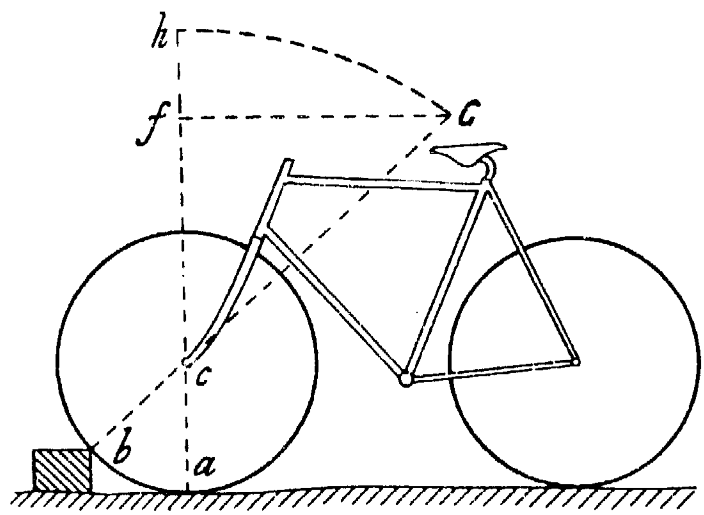


FIG. 200.

(II) *Brake on Back Wheel.*—If the brake be applied to the rear, instead of the front wheel, the bicycle is much safer as regards headers. If the brake, in this case, be applied too suddenly, the retarding force causes an incipient header, the frame turning about the front wheel centre c as axis, and the rear wheel immediately rises slightly from the ground. The retarding force being thus removed, the development of the header is arrested, the rear wheel again falls to the ground, and the process is repeated, a kind of equilibrium being established.

Headers through Obstructions on the Road.—If the check to the progress of the machine be caused by an obstruction on the road, the only difference from the case treated above is that the front wheel is free to revolve in its bearings; the header is taken about the point c as a centre, and the resultant R of the weight W and the force F_3 must not pass in front of the wheel centre c.

The direction of the forces between two bodies in contact is (neglecting friction) at right angles to the surface of contact. In a bicycle wheel with no friction at the hub, the direction of the pressure exerted by a stone at the rim must therefore pass through the wheel centre. This condition enables us to determine the size of the largest stone which can be ridden over at high speed without causing a header. Join the mass-centre, *G*, to the front wheel centre, *c* (fig. 201), and produce the line to cut the circum-

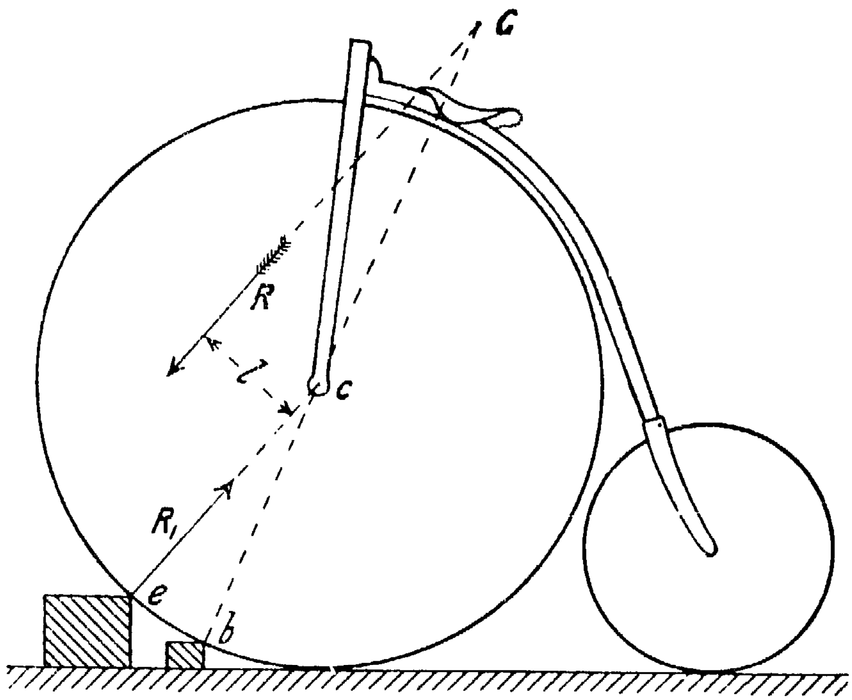


FIG. 201.

ference of the wheel at *b*. A stone touching the rim at a point higher than *b* may cause a header at high speed; a stone touching at a lower point may be ridden over at any speed. Figure 200 is the same diagram for a 'Safety' bicycle, a glance at which shows that with this machine a much larger stone can be safely surmounted than with an 'Ordinary.'

The above discussion presupposes that at the instant the front wheel strikes the stone no driving force is being exerted. If the rider is driving the front wheel forward at the instant, a larger obstacle may be safely surmounted. Let *e* (fig. 201) be the point of contact of a large stone; the reaction R_1 is in the direction *e c*. The resultant force *R* on the mass at *G* must be equal and parallel but opposite to R_1. The forces *R* and R_1 form a

couple $R\,l$, tending to turn the frame and rider about the centre c, l being the length of the perpendicular from G on $e\,c$ produced. If the rider apply to the front wheel a turning moment in the forward direction equal to or greater than $R\,l$, there will be a couple of equal magnitude acting on the frame tending to turn it in the opposite direction, which will neutralise the couple $R\,l$. The final result is that the wheel safely surmounts the obstacle, turning about e as centre.

CHAPTER XVIII

STEERING OF CYCLES

174. **Steering in General.**—When a bicycle moves in a straight line, the axes of its wheels are parallel to each other. The steering is effected by changing the direction of one of the wheel spindles relatively to the other. In order to effect this change of direction, the frame carrying the wheels is made in two parts, jointed to each other at the *steering-head*, the parts being called respectively the rear- and front-frames. One of these parts, that carrying the saddle, is usually much larger than the other (and is often called the frame, to the exclusion of the other part called *the fork*); the wheel—or wheels—mounted on the other (smaller) part of the frame is called the steering-wheel—or wheels. According to this definition, the driving-wheel of an 'Ordinary' is also the steering-wheel. In side-steering tricycles (see chap. xvi.) the frame is in three parts, and there are two steering-heads.

Cycles are *front-* or *rear-*steerers, according as the steering-wheel is mounted on the front- or rear-frame. All bicycles that have attained to any degree of public favour are front-steerers: The 'Ordinary,' the 'Kangaroo,' the 'Rover Safety,' the 'American Star,' and the 'Geared Ordinary.' A few successful tricycles have, however, been rear-steerers.

175. **Bicycle Steering.**—Let *a* (fig. 202) be the wheel fixed to the rear-frame, *b* the steering-wheel, and *d* the intersection of the steering-axis with the ground; this, in most cases, is at or near the point of contact of the wheel with the ground, though in the 'Rover Safety,' with straight front forks, it occurs some little distance in *front*. Let the plan of the axes of the wheels *a* and *b* be produced to meet at *o*, then if the wheels roll, without slipping sideways, on the ground, the bicycle must move in a circle having

o as its centre. The steering-wheel, *b*, will describe an arc of larger radius than that described by the wheel *a*; consequently if in making a sharp turn to avoid an obstacle the front wheel clears, the rear wheel will also clear. In a rear-steering bicycle, on the other hand, it may happen that the rear wheel may foul an object which has been cleared by the front wheel.

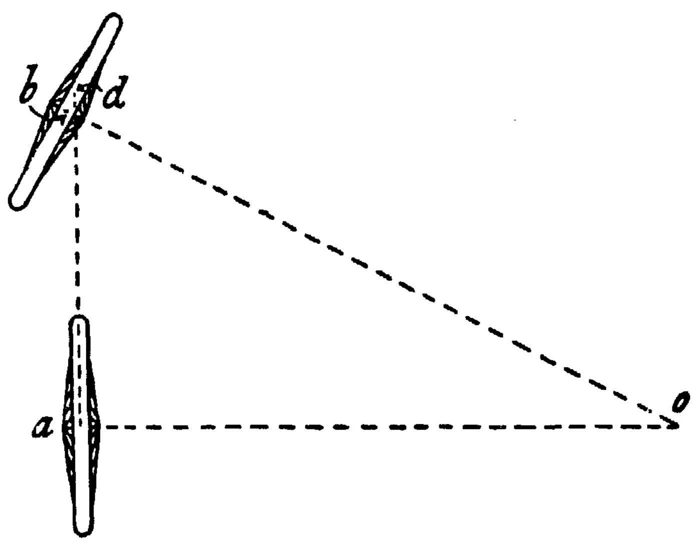


FIG. 202.

The actual sequence of operations in steering a bicycle is not commonly understood. If a beginner turn the steering-wheel to one side before his body and the bicycle have attained the necessary inclination, the balance will be lost. On the other hand, the beginner is often told to lean sideways in the direction he wants to steer. This operation cannot, however, be directly performed; since, if he lean his body to the right, the bicycle will lean to the left, and the sideway motion of the mass-centre cannot be controlled in this way. It has been shown (sec. 162) that the path described by a bicycle, even when being ridden as straight as possible, is made up of a series of curves, the bicycle being inclined alternately to the right and to the left. If at the instant of resolving to steer suddenly to one side the bicyclist be inclined to that side, he simply delays turning the steering-wheel until his inclination has become comparatively large. The radius of curvature of the path corresponding to the large inclination being small, the steering-wheel can then be turned, and the bicycle will describe a curve of short radius. If, on the other hand, he be inclined to the opposite side, the steering-wheel is at first turned in the direction opposite to that in which he wishes to steer, so as to bring the bicycle vertical, and then change its inclination; the further sequence of operations is the same as in the former case. Thus, to avoid an object it is often necessary to steer for a small fraction of a second towards it, then steer away from it; this

is probably the most difficult operation the beginner has to master. In steering, the rider's body should remain quite rigid in relation to the frame of the bicycle.

176. **Steering of Tricycles.**—The arrangement of the steering gear of a tricycle should be such that in rounding a corner the axes of the three wheels all intersect at the same point. In the 'Humber,' the 'Cripper,' and any tricycle with a pair of wheels mounted on one axle this condition is satisfied.

Let *O* be the intersection of the axes, *a*, *b*, *c*, of the three wheels. The tricycle as a whole rotating round *O* as a centre, the

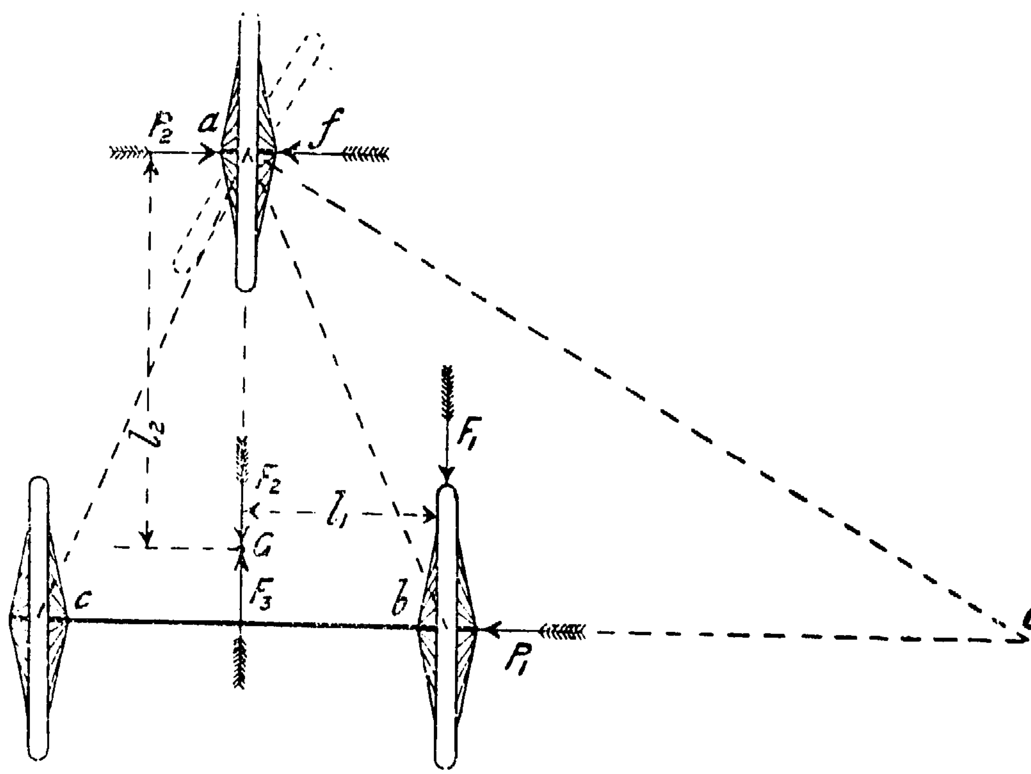


FIG. 203.

linear speed of the rim of wheel *c* will be greater than that of wheel *b* nearer the centre of rotation. If *b* and *c* are not driving-wheels, and are mounted independently on the axle, they will run automatically at the proper speeds. If *b* and *c* are driving-wheels, as in the 'Humber,' 'Cripper,' and 'Invincible' tricycles, some provision must be made to allow the wheel on the outside of the curve to travel faster than the inner. This is described in sections 188, 189.

177. **Weight on Steering-wheel.**—We have already seen that a considerable portion of the total weight of the machine must be

placed on the driving-wheel, so as to prevent skidding under the action of the driving effort. A certain amount of weight must also rest on the steering-wheel in order that it may perform its functions properly.

If the machine be moving at a high speed in a curve of short radius, the motion of the frame and rider can be expressed either as one of rotation about the point O, or as a translation equal to that of the mass-centre of the machine and rider, combined with a rotation about a vertical axis through the mass-centre G. If the rider should want to change from a straight to a curved course, the linear motion of the machine remains the same, but a rotation about an axis through the mass-centre must be impressed on it. To produce this a couple must act on the machine. The external forces, P_1 and P_2, constituting this couple can evidently only act at the points of contact of the wheel and the ground, and, presuming that the rolling friction may be neglected, can only be at right angles to the direction of rolling. The magnitudes of the forces P_1 and P_2 depend on the speed at which the cycle is running, and also on the general distribution of weight of the machine and rider—in mathematical language, on the moment of inertia of the system. The weight, w, on the steering-wheel must be equal to, or greater than, $\frac{P_1}{\mu}$, μ being the coefficient of friction. The moment of inertia, about its mass-centre, of a system consisting of a machine and two riders is very much greater than twice that of a system consisting of a machine and one rider; consequently the pressure required on the steering-wheels of tandems is much greater than twice that required on the steering-wheel of a single machine.

A simple analogy may help towards a better understanding of this. Suppose two persons of equal weight be seated at opposite ends of a see-saw, and that the up-and-down motion is imparted by a person standing on the ground, and applying force at one end of the see-saw. If now only one person be left on the see-saw, and he be placed at the middle exactly over the support, the person standing on the ground will have to supply a much smaller force than in the former case to produce swings of equal speed and amplitude. The swinging up and down of the see-saw corresponds

to the change of steering of the cycle from left to right, the forces applied by the person standing on the ground to the forces, P_1 and P_2, of reaction of the ground on the wheels. The single person on the middle of the see-saw corresponds to a single rider on a cycle, the two persons at the ends to the riders on a tandem.

Sensitiveness of Steering.—We have continually spoken of the point of contact of a wheel with the ground, thereby meaning the geometrical point of contact of a circle of diameter equal to that of the wheel. The actual contact of a wheel with the ground takes place over a considerable surface, the lower portion of the tyre getting flattened out as shown, somewhat exaggerated in figure 204. The total pressure of the wheel on the ground is distributed over this area of contact.

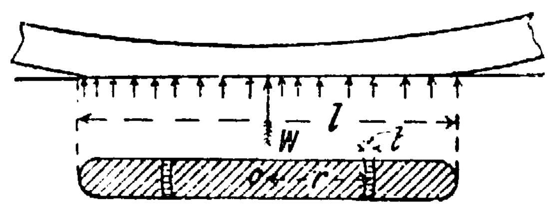


FIG. 204.

Considering tyres of the same thickness, it is evident that a wheel of large diameter will have the length of its surface of contact in the direction of the plane of the wheel greater than that of a wheel of smaller diameter.

Consider now the resistance to turning such a wheel, pivot-like, on the ground, as must be done in steering. Let A be the area of the surface of contact, and suppose the pressure of intensity, p, distributed uniformly over it, as will be very approximately the case with pneumatic tyres; then

$$p = \frac{W}{A}.$$

Consider a small portion of the area of width, t, included between two concentric circular arcs of mean radius, r. Let a be the area of this piece, the total pressure on this will be $p\,a$, and the frictional resistance to spinning motion of this portion of the tyre on the ground will be $\mu\,p\,a$. The moment of this force about the geometrical centre, O, is

$$\mu\,p\,a\,r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and the total moment of resistance of the wheel to spinning on the ground is the sum of all such elements. If we consider the surface of contact to be a narrow rectangle, whose width is very

small in comparison with its length, l, the *average* value of r in (1) will be $\frac{l}{4}$, and the total moment of resistance to spinning will be

$$\frac{\mu W l}{4} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Thus a greater pull will be required at the handle-bar to steer a large wheel than a small one; in other words, a small steering-wheel is more sensitive than a large one. The assumption made above, that the width of the surface of contact is very small compared with its length, is not even approximately true for pneumatic tyres. The moment of resistance in this case will, however, increase with l, and, therefore, the conclusion as to the relative sensitiveness of small and large wheels holds.

The above expression gives the moment of resistance to turning the steering-wheel on the ground when the bicycle is at rest. This moment is quite considerable, and is much greater than the actual moment required to steer when the bicycle is in motion, as can be easily verified by experiment. The explanation of this phenomenon is practically of the same nature as the explanation, given in section 170, of the small force necessary to overcome friction in one direction, provided motion in a direction at right angles exists. In the present case the wheel is rotating about a horizontal axis during its forward motion; the steering is effected by giving it a motion about a vertical axis. On account of the motion about a horizontal axis already existing, a comparatively small moment is sufficient to overcome the frictional resistance to motion about a vertical axis.

178. **Motion of Cycle Wheel.**—It is a popular notion that the motion of a vehicle wheel is one of pure rolling on the ground, but a little consideration will show that this is not always the case. So long as a tricycle moves in a straight line, the wheels merely roll on the ground, the instantaneous axis of rotation being a line through the point of contact of the wheel and ground, parallel to the axis. When the vehicle is moving in a curve, in addition to this rotation about a horizontal axis, the wheel possesses a motion round a vertical axis, and some parts of the tyre in contact with the ground slide over the ground, as described in section 177. The

instantaneous axis of rotation is now a line inclined to the ground.

Suppose that the plane of the wheel can be inclined to the vertical when the cycle is moving in a curve, as in the case of a bicycle or steering-wheel of a 'Cripper' tricycle. Let the axis of the wheel be produced to cut the ground at V, then if the cycle be at the instant turning about the point V as centre, the motion of the wheel on the ground will be one of pure rolling, no sliding being experienced by any point of the tyre in contact with the ground. The part of the wheel in contact with the ground may be considered part of a right circular cone, having its vertex at V. Such a cone would roll without slipping on a plane surface, the vertex, V, of the cone remaining always in the same position.

The intersection of the axis of the wheel with the ground is determined by the inclination of the wheel to the vertical. This inclination depends on the radius of the curve in which the bicycle is moving, and also its speed. For a curve of a given radius there is, therefore, one particular speed at which V will coincide with O, the centre of turning of the bicycle. At this speed there will be no spinning of the tyre on the ground, while at greater or less speeds spinning occurs to a greater or less degree.

179. **Steering Without Hands.** In a front-driving bicycle, the saddle and crank-axle being carried by the rear- and front-frames respectively, there is theoretically no difficulty in steering without using the handle-bar. If it be desired to turn towards the right, a horizontal thrust at the left pedal as it passes its top position, or a pull at the right pedal as it passes its lowest position, will effect the desired motion.

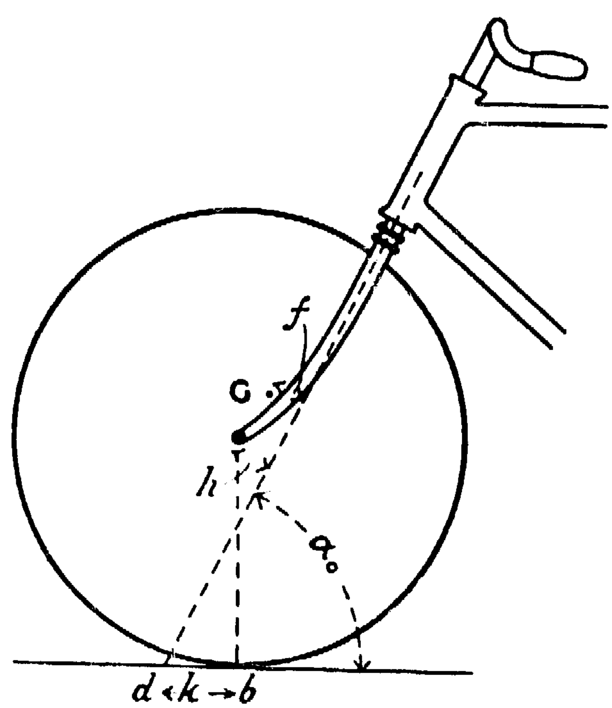


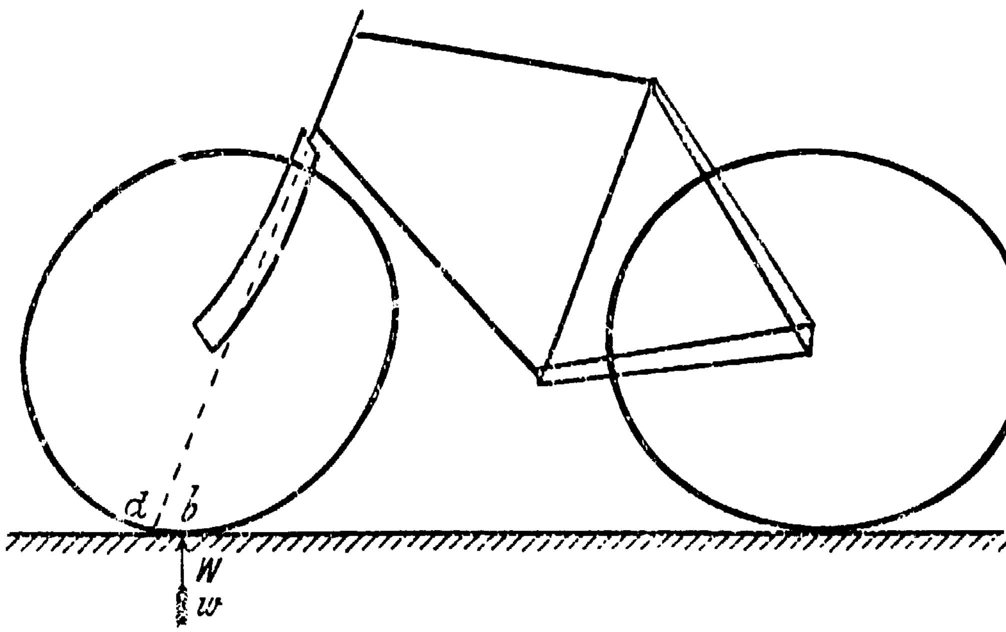
FIG. 205.

In a rear-driving bicycle, the saddle and crank-axle being

carried by the rear-frame, there is no direct connection between the rider and the steering-wheel axle except by the handle-bar.

Let α_0 be the angle the steering-axis makes with the horizontal when the bicycle is vertical (fig. 205); h the distance of the wheel centre from the steering-axis; k, the distance between b, the point of contact of the wheel with the ground, and d the point of intersection of the steering-axis with the ground, when the bicycle is vertical and the steering-wheel in its middle position; f the

FIG. 206.



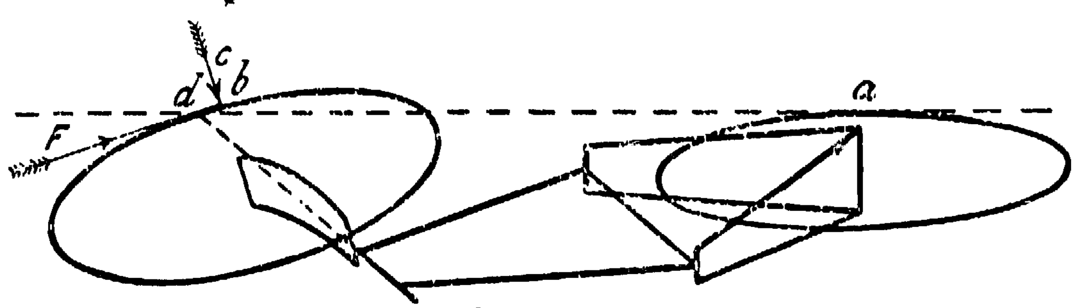
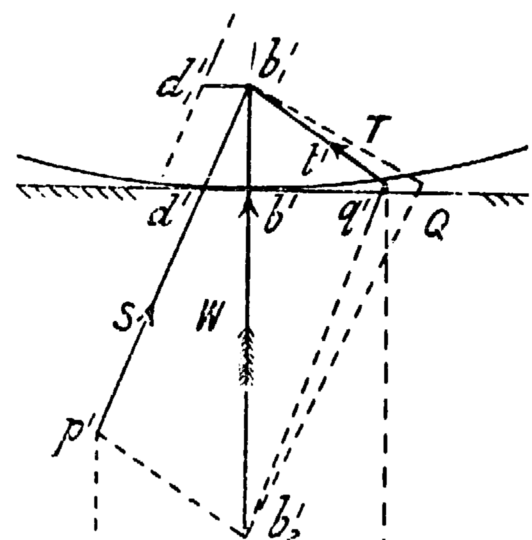


FIG. 207.

distance of the mass-centre of the steering-wheel and front-frame (including handle-bar, &c.) from the steering-axis; θ the inclination of the middle plane of the rear-frame to the vertical; ϕ the angle the handle-bar is moved from its middle position, *i.e.* the angle between the middle planes of the front and rear wheels; and α the angle the steering-axis makes with the horizontal, corresponding to the values of θ and ϕ. Figs. 206 and 207 are elevation and plan of a bicycle heeling over. The forces acting on the front wheel and frame which may tend to turn it about the

steering-axis are—the reaction of the ground, and the weight, w, of the front wheel and frame. The reactions at the ball-head intersect the steering-axis, and therefore cause no tendency to turn. The reaction of the ground can be resolved into three components—W, acting vertically upwards; F, the resistance in the direction of motion of the wheel; and C, the centripetal force at right angles to F. The line of action of F passes very near the steering-axis for all values of θ and ϕ, and since F is itself small in comparison with W and C, its moment may be neglected. Figs. 208 and 209 are elevation and plan enlarged from figs. 206 and 207, showing the relation of W to the steering-axis. $b\ d_1$ is the plan and $b_1^1\ d_1^1$ the elevation of the shortest line between W and the steering-axis. W can be resolved into a force, S, parallel to the steering-axis, and a force, T, at right angles to the plane containing S and the steering-axis. If $b_2^1\ b_1^1$ represent W to scale, $q^1\ b_1^1$ and $p^1\ b_1^1$ are the elevations of the forces T and S, while $Q\ b_1^1$ and $b_2^1\ Q$ show to scale the true magnitudes of T and S respectively; *i.e.* $b_1^1\ b_2^1\ q^1$ is the elevation of the force-triangle, and $b_1^1\ b_2^1\ Q$ is its true shape. Also it may be noticed that the line $b\ d_1$ in plan measures the true length of the perpendicular between W and the steering-axis; and W tends to turn the steering-wheel still further, its moment about the steering-axis being $\overline{Q\ b_1^1} \times \overline{b\ d_1}$. The centripetal force C tends to turn the steering-wheel back into its middle position. The effect of the weight w in tending to turn the steering-wheel can be shown in exactly the same way as that of the vertical reaction W. The tendency is in general to increase the deviation of the steering-wheel, but when a straight fork is used the

FIG. 208.



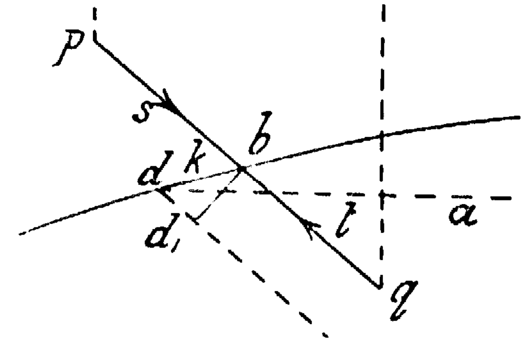


FIG. 209.

tendency is to reduce it, on account of the mass-centre of the handles being behind the steering-axis.

We shall now determine the analytical expressions for the moments of W, C, and w, assuming that the angles θ and ϕ are small, and that, therefore, we may use the approximations

$$\sin\theta = \theta = \tan\theta$$
$$\sin\phi = \phi = \tan\phi.$$

We have seen above that the moment of W is

$$\overline{Q\,b_1{}^1} \times \overline{b\,d_1}.$$

Now $\qquad \overline{Q\,b_1{}^1} = W\cos\alpha,$

also $\qquad \sin\alpha = \sin\alpha_0\cos\theta.$

Therefore $\qquad \overline{Q\,b_1{}^1} = W\sqrt{1 - \sin^2\alpha\cos^2\theta}$

$$= W\cos\alpha_0 \text{ approximately.}$$

Now $\overline{b\,d_1} = \overline{b\,d}\sin b\,d\,d_1$. The angle $b\,d\,d_1$ is made up of the two angles $a\,d\,b$ and $a\,d\,d_1$. The former is zero if ϕ is zero, and the latter is zero if θ is zero. For small values of θ and ϕ, the angle $a\,d\,b = \phi\sin\alpha_0$, and $a\,d\,d_1 = \theta\tan\alpha_0$.

Therefore $\qquad \overline{b\,d_1} = \overline{b\,d}\sin(\theta\tan\alpha_0 + \phi\sin\alpha_0).$

Therefore, if we assume that $\overline{b\,d}$ remains constant, we have $\overline{b\,d_1} = k(\theta\tan\alpha_0 + \phi\sin\alpha_0)$ approximately, and moment of W is

$$W k \sin\alpha_0 (\theta + \phi\cos\alpha_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The moment of C for small values of θ and ϕ will be approximately $C \times b\,d \times \sin\alpha_0$.

Now, if the angle ϕ remains constant

$$C = \frac{W v^2}{g R},\quad R = \frac{l}{\sin a\,d\,b} = \frac{l}{\phi\sin\alpha_0} \text{ approximately,}$$

v being the speed of the bicycle, R the radius of the circle described by the front wheel, and l the length of the wheel-base. Therefore the moment of C is

$$\frac{W v^2 k\,\phi\sin^2\alpha_0}{g\,l} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The moment of w can be found as follows: Resolving w into two components parallel to and at right angles to the steering-axis, the latter is $w \cos \alpha$. Figure 210 shows side and end elevations of the steering-axis and mass-centre, G. The perpendicular distance $B_1 B_2$ between w and the steering-axis for a small value of θ is

$$\overline{G B_2} \times \theta = \frac{f \theta}{\cos \alpha_0},$$

while for a small value of ϕ it is $f \phi$. Therefore moment of w is

$$w \cos \alpha_0 \left(\frac{f \theta}{\cos \alpha_0} + f \phi \right)$$

$$= w f (\theta + \phi \cos \alpha_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Hence, finally adding (3), (4), and (5), the moment tending to turn the steering-wheel still further from its middle position is

$$W k \sin \alpha_0 (\vartheta + \phi \cos \alpha_0) - \frac{W k \phi v^2 \sin^2 \alpha_0}{g l} + w f (\theta + \phi \cos \alpha_0)$$

$$= (W k \sin \alpha_0 + w f) (\vartheta - \phi \cos \alpha_0) - \frac{W k \phi \sin^2 \alpha_0}{g l} v^2 \quad . \quad (6)$$

To maintain equilibrium the expression (6) should have the value zero, to steer further to one side or other it should have a small positive value, and to steer straighter a small negative value.

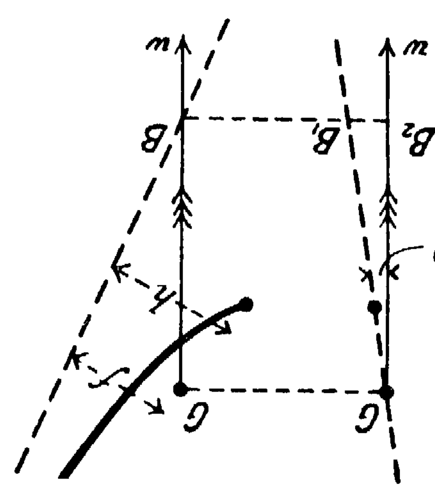


FIG. 210.

For given values of v and ϕ there remains an element θ, the inclination of the rear-frame, at the command of the rider; but even with a skilled rider the above moment varies probably so quickly that he could not adjust the inclination θ quickly enough to preserve equilibrium.

In the above expressions we have taken no account of the gyroscopic action of the wheel, though probably this is the most important factor in the problem Taking account of the gyroscopic action, the above moment about the steering-axis would produce a motion of precession about an axis at right angles to those of the ball-head and

steering-wheel ; while to turn the steering-wheel about the steering-axis, a couple, with its axis at right angles to the steering-axis, would be required. This is produced by the side pressures on the steering tube; so that in steering without hands, if the rider wishes to turn to the right, he merely leans over slightly to the right, and the steering-wheel receives the required motion, provided the value of the expression (6) is small.

Example.—With the same data as in section 168, to turn the steering-wheel at the speed indicated, a couple of 2·42 foot-lbs. is required, *i.e.* if the ball-head be 8 inches long, side pressures of 3·63 lbs. would suffice to turn the front wheel at the speed indicated. To turn the steering-wheel more quickly, a greater side pressure must be exerted on the steering-head.

From section 168 the gyroscopic couple required is proportional to the square of the speed, and approximately proportional to the weight and to the diameter of the front wheel ; therefore, steering without hands should be easier the higher the speed, the larger the steering-wheel, and the heavier the rim of the steering-wheel. This agrees with the fact that a fair speed is necessary to perform the feat, that the feat is easier with pneumatic than with solid tyres, the former with rim being heavier than the latter ; it also accounts for the *easy* steering with large front wheels, and for the fact that the 'Bantam' is more difficult to steer without hands than the 'Ordinary.'

It may be noticed that if this explanation be correct, it should be possible to ride without hands a bicycle in which the steering-axis cuts the ground at the point of contact of the front wheel. M. Bourlet, who discusses the subject at considerable length, says this is impossible ; he also says that the mass-centre of the front wheel and frame must lie in front of the steering-axis ; but this would mean that a bicycle with straight forks could not be ridden without hands ; whereas some of the earliest 'Safety' bicycles, made with straight forks, were easily ridden without hands.

180. **Tendency of an Obstacle on the Road to Cause Swerving.**—If a bicycle run over a stone, the force exerted by the stone on the steering-wheel acts in a direction intersecting the steering-axis, and has thus no tendency to cause the steering-wheel to turn in either direction. In the same way, the steering-wheel of a

'Cripper' or 'Invincible' tricycle in running over a stone experiences no tendency to turn, and therefore no resistance need be applied by the rider at the handle-bar. The line of action of the force exerted on the machine cuts a vertical line through the mass-centre ; the force therefore only tends to reduce the speed of the machine, but not to deviate it from its path. If the obstacle meet one of the side wheels of a tricycle, the force exerted by the stone and the force of inertia of the rider form a couple tending to turn the machine and rider as a whole about their common mass-centre. In some tricycles the force exerted by the stone tends also to change the position of the steering gear, and so cause sudden swerving. A few of the chief types of tricycles are discussed in detail, with reference to these points, in the following sections.

181. **Cripper Tricycle** — Let one of the driving-wheels meet with an obstacle. Introducing at G, the mass-centre, two opposite forces, F_2 and F_3, each equal to F_1, no change is made in the static condition of the system. The force, F_1 (fig. 203), exerted by the stone on the machine is equivalent to an equal force, F_2, acting at the mass-centre of the machine and rider, and retarding the motion, and a couple formed by the forces F_1 and F_3 tending to turn the machine about its mass-centre, G. This turning is prevented by the side friction of the wheels on the ground. To actually turn about G, the driving-wheels must roll a little and the front steering-wheel slip sideways.

Let f be the resistance to slipping sideways of the front wheel, l_1 and l_2 the lengths of the perpendiculars from G on the lines of action of the forces F_1 and f, w the load on the steering-wheel, and μ the coefficient of friction between the steering-wheel and the ground. Then $f l_2$ must be equal to or greater than $F_1 l_1$. Also $f = \mu w$, therefore $\mu w l_2 \geq F l$, or

$$w \geq \frac{F l_1}{\mu l_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

If, in the 'Cripper' tricycle, the steering-axis produced passes exactly through the point of contact of the steering-wheel with the ground (fig. 211), the reaction from the ground on the steering-wheel has no tendency to cause it to turn ; no resistance is necessary

at the handle-bar when one of the driving-wheels strikes an obstacle. If, as in all modern tricycles, the steering-axis produced passes in front of the point of contact of the steering-wheel with the ground (fig. 212), the force, f, will tend to turn the steering-wheel sideways, and must be resisted by a force, F_4, at the handle-bar, such that $F_4 l_4 = f l_3$, l_3 being the length of the perpendicular

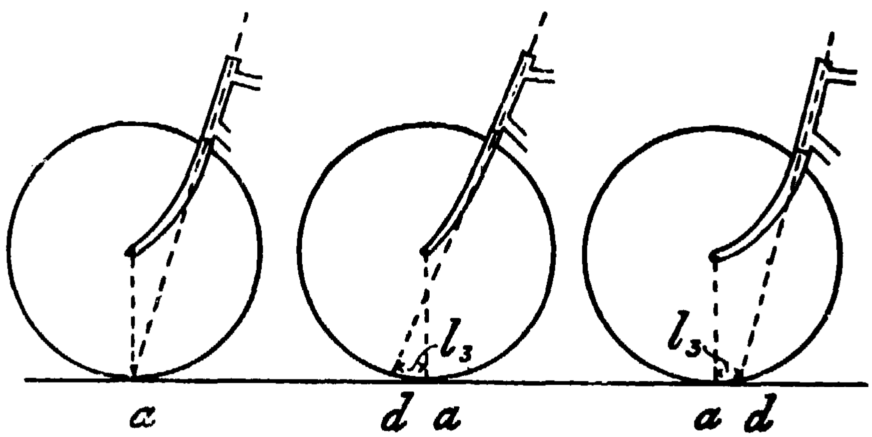


FIG. 211. FIG. 212. FIG. 213.

from the point of contact with the ground to the steering-axis, and l_4 the half-length of the handle-bar.

In a tricycle with a straight fork, the distance l_3, and therefore also the necessary force F_4, at the handle-bar to prevent swerving, is greater than with a curved fork (fig. 212).

182. **Royal Crescent Tricycle.**—In the 'Royal Crescent' tricycle (fig. 151), made by Messrs. Rudge & Co., the steering-axis intersected the ground at a point d (fig. 213), some distance behind the point of contact of the wheel. The force, f, would therefore tend to turn the steering-wheel about the steering-axis, in the opposite direction to that in the 'Cripper.' The distance, l_3, being much greater than in the 'Cripper,' the force, F_4, necessary at the handle-bar to prevent swerving was also greater. A spring control was used for the steering, so that a considerable force was necessary to move the steering-wheel from its middle position.

183. **Humber Tricycle.**—In a 'Humber' tricycle, an obstacle in front of one of the driving-wheels tends to turn the driving-axle round the steering-axis, a (fig. 214). This must be resisted by a force, F_4, applied by the rider at the handle-bar given by the equation $F_1 l_1 = F_4 l_4$, or the obstacle will change the direction of motion suddenly and a spill may occur. If the rider supply the

necessary force, F_4, the conditions as to the machine as a whole turning about the mass-centre G, and as to the weight necessary on the steering-wheel to prevent this turning, are the same as discussed in section 181.

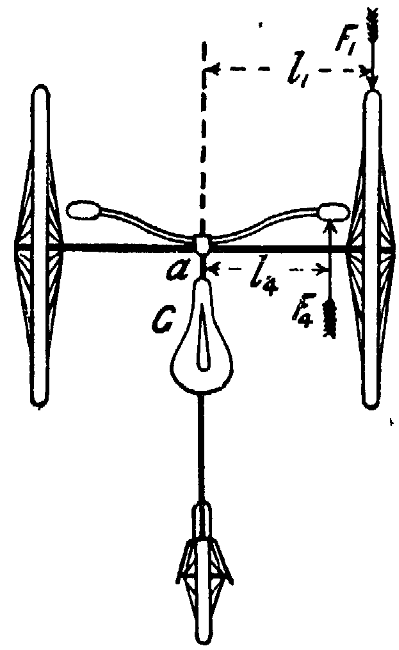


FIG. 214.

It will be seen from the above that the arrangement of the steering in the 'Humber' tricycle is less satisfactory than in some of the other types.

Any cycle in which there are a pair of independent wheels mounted on a common axle, pivoted to the frame at its middle point, will be subject to the same defect of steering. Examples are afforded in figures 154, 155, and 182.

184. **Olympia Tricycle and Rudge Quadricycle.**—The wheel plan of an 'Olympia' tricycle is shown at figure 215. A single rear driving-wheel is used; the two front wheels are side-steerers. In some of the earlier patterns of this tricycle made by Marriott &

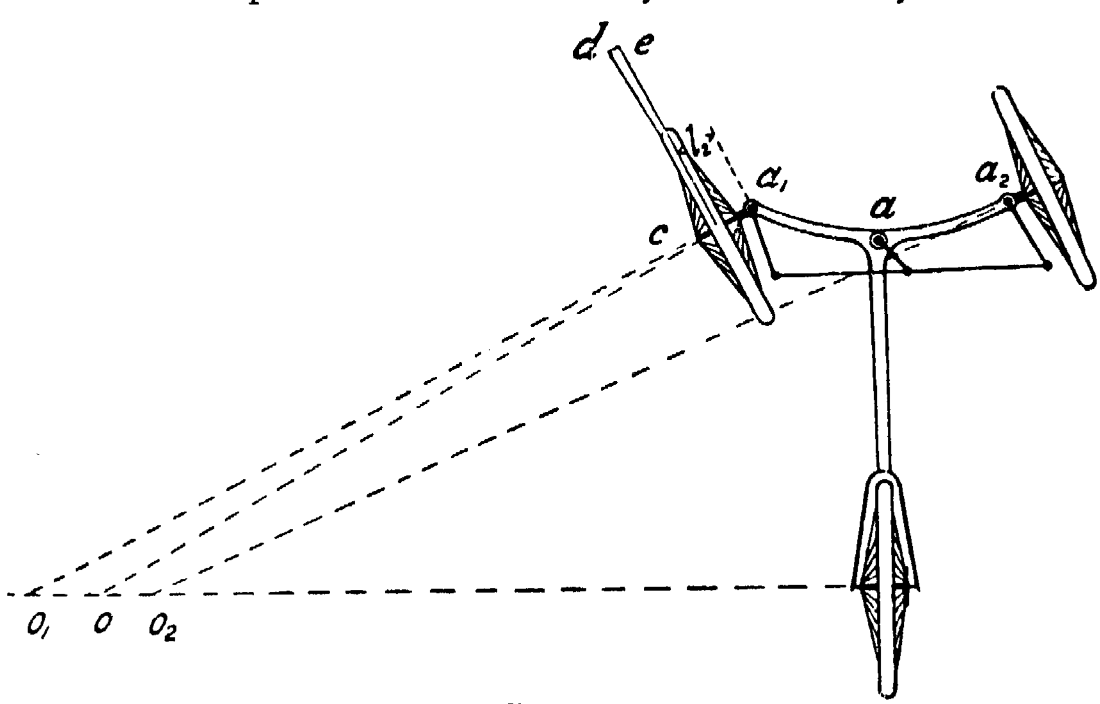


FIG. 215.

Cooper, the steering-wheels ran free on the same axle, which was pivoted at a to the rear-frame of the machine; the action in steering was therefore the same as in the 'Humber' tricycle. In

the modern patterns of the 'Olympia' tricycle the steering is effected by providing the steering-wheel spindles with separate steering-heads at a_1 and a_2. Short bell-cranks are formed on the spindles, and the ends of these cranks are connected by links to the end of a crank at the bottom of the steering-post a. The distance, l_2, between the steering-axis and the point of contact of the steering-wheel with the ground being much less than in the 'Humber' tricycle, the influence of an obstacle in causing swerving is correspondingly less, though in this respect the 'Olympia' is inferior to the 'Cripper.' The arrangement of this gear should be such that the axes of the steering-wheels in any position intersect at a point, O, situated somewhere on the axis of the driving-wheel. This cannot possibly be effected by any arrangement of linkwork, but the approximation to exactness may be practically all that can be desired for road riding. The gear should be arranged so that the bell-crank of the outer steering-wheel swings through a less angle from its middle position than that of the inner wheel.

If the axes of the wheels a_1 and a_2 intersect the axis of the driving-wheel at O_1 and O_2 (fig. 215), the machine as a whole may be supposed to turn about a point, O, somewhere between O_1 and O_2. Let c be the point of contact of wheel a_1 with the ground when the tricycle is moving round centre O, and let the linear velocity of a point on the frame vertically above c be represented by $c\,d$, drawn perpendicular to $O\,c$. From c draw $c\,e$ perpendicular, and from d draw $d\,e$ parallel, to the axis $O_1\,c$; these two lines intersecting at e, the actual velocity $c\,d$ is compounded of a velocity of rolling $c\,e$ of the wheel on the ground, and a velocity of side-slip, $e\,d$. The existence of this side-slip in running round curves necessitates careful arrangement of the steering mechanism, so that the centres O_1 and O_2 may never be widely separate. This side-slip must also add appreciably to the effort required to propel the 'Olympia' tricycle in a curved path, such as a racing track; and for such a purpose might possibly appreciably handicap it as compared with a 'Cripper.'

The steering gear of the 'Rudge' quadricycle is the same as that of the 'Olympia' tricycle.

185. **Rudge Coventry Rotary.**—In the 'Rudge Coventry Rotary' two-track tricycle, with single driving-wheel and two

steering-wheels (fig. 216), the reaction from the ground in driving being at F, there was continually a couple, $F l_1$, in action tending to turn the machine, and which was resisted by the reactions, f_1 and f_2, of the ground on the sides of the two side wheels. For equilibrium,

$$F l_1 = f l_2.$$

The steering-wheels were pivoted about axes passing through their points of contact with the ground and connected by short levers, connecting-rods, and a toothed-rack, to a toothed-wheel controlled by the rider. The arrangement, in this case, should again be such that in any position of the steering-gear the three axes intersect at a point O; the machine would then turn about O as a centre.

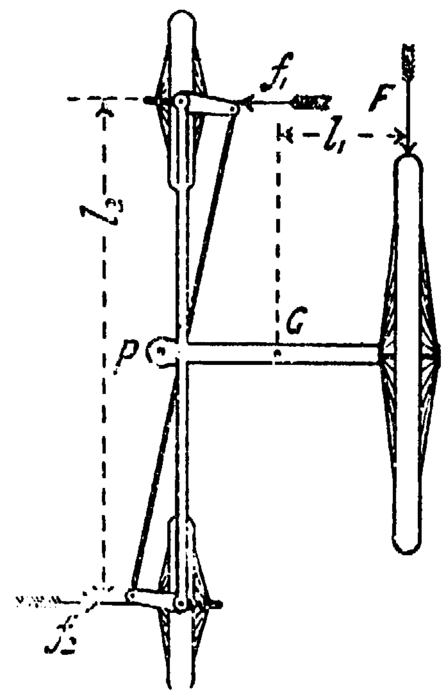


FIG. 216.

If either of the steering-wheels pass over an obstacle, it is evident that since the direction of the force acting on the wheel intersects the steering-axis there will be no tendency to turn the wheel, and therefore no resistance need be offered at the handle by the rider. The tendency of an obstacle to turn the machine as a whole about the mass-centre, G, is discussed in exactly the same way as for the 'Cripper' tricycle.

186. **Otto Dicycle.**—In the 'Otto' dicycle, the steering was effected by connecting each of the driving-wheels, by means of a smooth pulley and steel band, to the crank-axle. To run round a corner, the tension on one of the bands was reduced by the motion of the steering-handle, the band slipped on its pulley, and the other wheel being driven at a faster rate, the machine described the curve required. In a newer pattern with central gear (fig. 172) the motion was transmitted by a chain from the crank-axle to the common axle of the two wheels. The wheel-axle was divided into two portions, a differential gear being used, as explained in section 189. In steering, one of the driving-wheels was partially braked by a leather-lined metal strap, thereby making it more difficult to run than the other wheel; one wheel was

thus driven faster than the other, and the machine described a curve.

If an obstacle met one of the wheels, its tendency was to retard the machine and to make it turn about its mass-centre. In performing this motion of rotation, neither of the wheels slipped sideways, and therefore no resistance was offered to the swerving; consequently some other provision had to be made to prevent this motion. This was accomplished by locking the gear when running straight, so that the two driving-wheels were, for the time being, rigidly fixed to the axle, and ran at the same speed. If the horizontal force, F, actually caused the machine to swerve, one or other of the wheels actually slid on the ground. The frictional resistance to this sliding was $\frac{\mu W}{2}$, W being the weight of the machine and rider. If F was less than this, and the mechanism acted properly, the machine moved straight ahead over the obstacle.

187. **Single and Double-driving Tricycles.**—A tricycle, in which only one of the three wheels is driven, is said to be *single-driving*. The 'Rudge' two-track and the 'Olympia' are familiar examples. In single-driving tricycles the two idle wheels are supported independently, so that the three wheels have perfect freedom to rotate at different speeds.

If the two driving-wheels of a double-driving tricycle are (as is almost invariably the case) of the same diameter, while driving in a straight line they rotate at the same speed. They could, therefore, be rigidly fixed on the same axle, if only required to run straight; but in running round a curve the outer wheel must rotate faster than the inner, unless one or other of the wheels skid, as well as roll, on the ground. Some arrangement of mechanism must be used to render possible the driving of the two wheels at different speeds.

188. **Clutch Gear for Tricycle Axles.**—Besides the 'Otto' double-driving gear above described, two others, the clutch gear and the differential (or balance) gear, have been used to a considerable extent, though at present the differential gear is the only one used. In the 'Cheylesmore' clutch gear (fig. 217), made by the Coventry Machinists Co., Limited, a sprocket wheel, w, in the

form of a shallow box, was mounted loosely near each end of the pedal crank-axle, and was connected by a chain to the corresponding driving-wheel. A cam, *c*, was fixed near each end of the crank-axle, and between the cam and the inner surface of the wheel, *w*, four balls, *b*, were placed; the four spaces between the cam and the rim of the toothed-wheel being narrower at one end, and wider at the other, than the ball. In driving the axle in the direction of the arrow, the balls, *b*, were jammed between the wheel and the cam, the wheel consequently turned with the axle. If the axle were turned in the opposite direction, or if the wheel tended to move faster than the axle in the direction of the arrow, the balls, *b*, were liberated, and the cog-wheel revolved quite independently of the axle. While moving in a straight line both driving-wheels were driven; but when running in a curve the inner wheel was driven by the clutch, while the outer wheel running faster than the inner overran the axle and liberated the balls, the outer wheel being thus left quite free to revolve at the required speed.

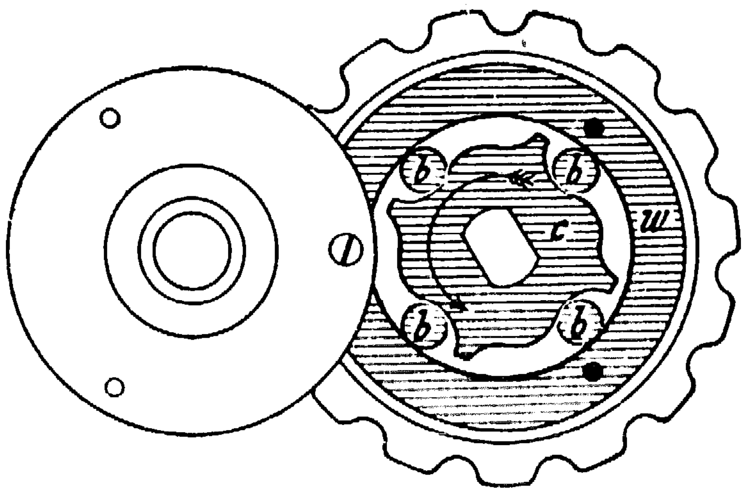


FIG. 217.

189. **Differential Gear for Tricycle Axle.**—Let two co-axial shafts, *m* and *n* (fig. 218), be geared to a shaft, *k*, the axis of which intersects that of the shafts, *m* and *n*, at right angles. The gearing may consist of three bevel wheels, *a*, *b*, and *c*, fixed respectively to shafts, *m*, *k*, and *n*. The three shafts are carried by bearings, m_1, k_1, and n_1 respectively. Let the shaft, *k*, be rotated in its bearings, it will communicate equal but opposite rotations to the shafts *m* and *n*. If ω_1 be the angular speed of the shaft *m*, that of *n* will be $-\omega_1$, and the relative angular speed of the shafts *m* and *n* will be $2\,\omega_1$.

Now, let the shaft, *k*, carrying with it its bearings, k_1, be rotated about the axis, *m n*, with an angular speed, ω; the teeth of the

wheel, b, engaging with those of a and c, will cause the shafts, m and n, to rotate with the same speed, ω, about their common axis ; the shaft, k, being at rest relative to its bearings, k_1. If driving-wheels be mounted at the ends of the shafts, m and n, they will both be driven with the same angular speed ω about the axis $m\,n$.

Let now the shaft, k, be rotated in its bearings, giving a rotation ω_1 to the shaft m, and a rotation $-\omega_1$ to the shaft n, while k and its bearings are being simultaneously rotated about the axis $m\,n$

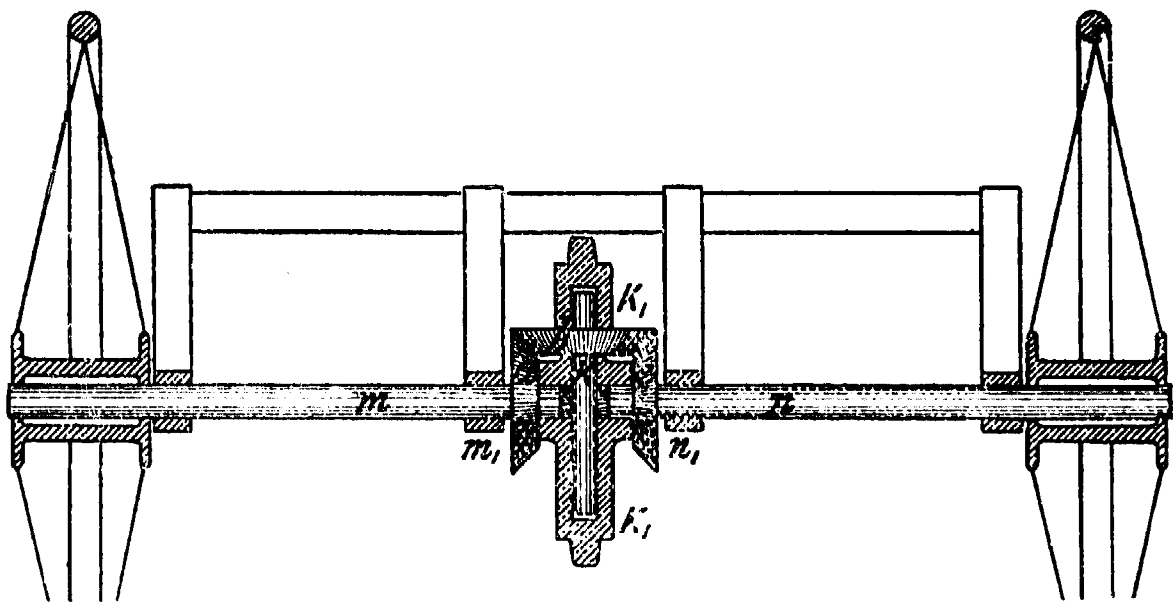


FIG. 218.

with the angular speed, ω. The resultant speed of the shaft m will be $(\omega+\omega_1)$, that of the shaft n will be $(\omega-\omega_1)$. Thus, finally, the average angular speed of the shafts m and n is the same as that of the bearings, k_1, while the difference of their angular speeds is quite independent of the angular speed of k_1. In Starley's *differential* tricycle gear, or *balance* gear, a chain-wheel is formed on the same piece of metal as the bearings, k_1, and is driven by a chain from the crank-axle. The driving effort of the rider is thus transmitted to the driving-wheels at the end of the shafts m and n. The shafts have still perfect freedom to rotate relatively to each other, and thus if in steering one wheel tends to go faster or slower than the other, there is nothing in the mechanism to prevent it.

In figure 218, the bevel-wheels, a and c, in gear with the wheel b are shown of equal size. In Starley's gear (fig. 219) a second wheel

near the other end of the spindle, k, gears with those on the ends of the two half axles, so that the driving effort is transmitted at two points to each of these wheels. This forms, perhaps, the neatest possible gear, but a great variety could be made if necessary. Such a differential gear consists essentially of the chain-wheel, k_1, carrying a shaft, k, which gears in any manner with the shafts m and n. The particular form of gearing is optional; provided that it allows m and n to rotate relatively to each other. Thus in Singer's double-driving gear, the wheel, b, was a spur pinion, with

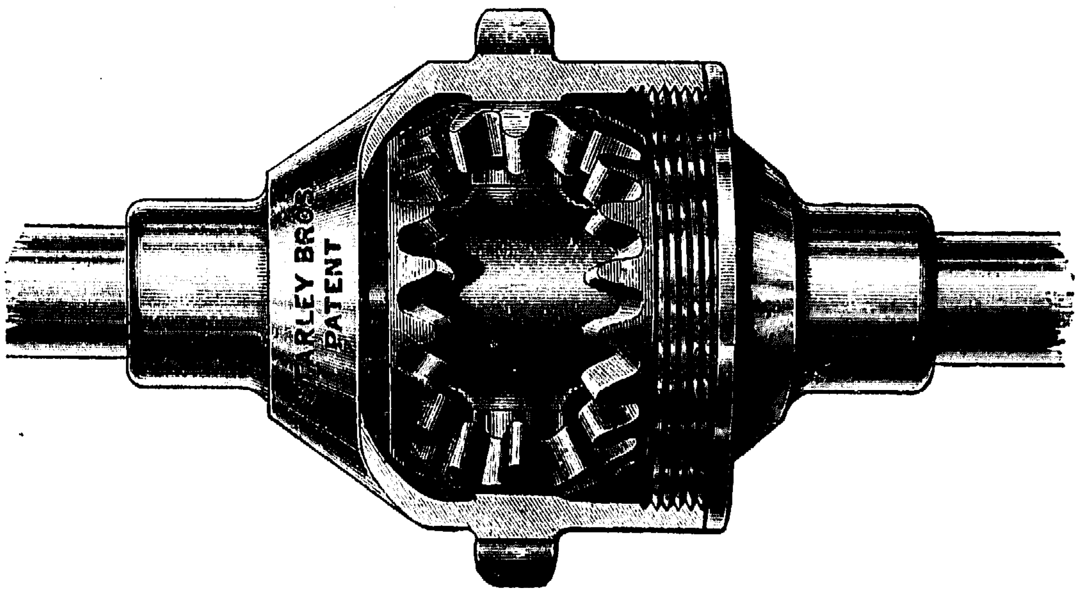


FIG. 219.

its axis parallel to m n, and engaging with a spur-wheel and an annular-wheel fixed respectively to the shafts, m and n. This gear had the slight disadvantage that equal efforts could not be communicated to the driving-wheels, that connected to the annular-wheel of the gear doing most of the work.

The balance gear being only used *differentially* for steering, the relative motion of the bevel-wheels, a, b, c (fig. 218), is very slow, and there is not the same absolute necessity for excessive accuracy as in toothed-wheel driving gear.

Example.—A tricycle with 28-in. driving-wheels, tracks 32 in. apart, being driven in a circle of 100 feet radius at a speed of 20 miles an hour, required the speed of the balance-gear.

While the centre of the machine moves in a circle 1200 inches radius, the inner and outer wheels move in circles $(1200 - 16)$ and $(1200 + 16)$ inches radii respectively. The circumferences of these

circles are respectively $2\pi \times 1200$, $2\pi \times 1184$, and $2\pi \times 1216$ inches. While the centre of the machine moves over $2\pi \times 1200$ inches, the outer wheel moves over $2\pi \times 32$ inches more than the inner. The relative linear speed is therefore $\frac{2\pi \times 32}{2\pi \times 1200} \times 20$

$$= \cdot 5333 \text{ miles per hour}$$

$$= \frac{\cdot 5333 \times 5280 \times 12}{60} = 563 \cdot 2 \text{ inches per minute.}$$

The circumference of a 28-in. wheel is 87·96 in. The number of revolutions made by the outer part of the axle in excess of those made by the inner is therefore

$$\frac{563 \cdot 2}{87 \cdot 96} = 6 \cdot 40 \text{ per minute.}$$

The number of revolutions of the axle divisions relative to the balance box, k, is therefore 3·20 per minute.

CHAPTER XIX

MOTION OVER UNEVEN SURFACES

190. **Motion over a Stone.**—If a cycle be moving along a perfectly smooth, flat road, neglecting the slight horizontal sideway motion due to steering, the motion of every part of the frame of the machine is in a straight line. Suppose a bicycle to move over a stone which is so narrow that its top may be considered a point. The motion being in the direction of the arrow, the path of the centre of the driving-wheel will be a straight line *O A* (fig. 220) parallel to the ground until the tyre

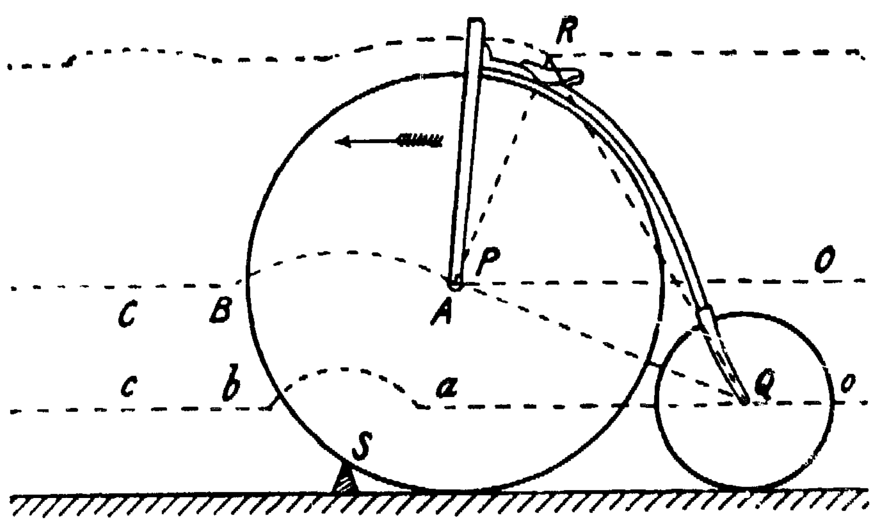


FIG. 220.

comes in contact with the obstacle at *S*, when the further motion of the wheel centre will be in a circular arc, *A B*, having *S* as centre. The further path of the wheel centre is the straight line, *B C*, parallel to the ground. The path of the centre of the rear wheel is of the same nature: a straight line, *o a*, until the tyre meets the obstacle *S*, the circular arc, *a b*, with *S* as centre, and then the straight line *b c*.

The motion of any point rigidly connected to the frame of

the bicycle can now be easily found. Let P and Q be the centres of the front and rear wheels respectively, and let it be required to find the form of the path of the point R lying on the saddle and rigidly connected to P and Q. Having drawn on the paper the paths of P and Q (fig. 220), take a small piece of tracing paper, and on it trace the triangle $P Q R$. Move this

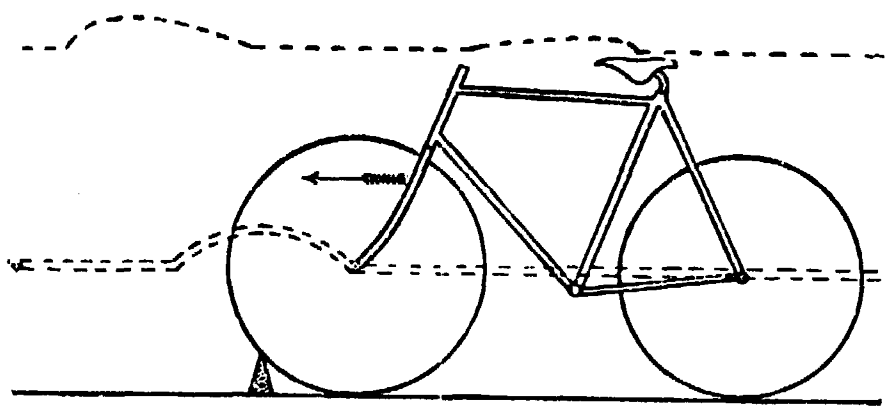


FIG. 221.

sheet of tracing paper over the drawing paper so that the points P and Q lie respectively on the curves $O A B C$ and $o a b c$. In this position prick through the point R, and a point on its path will be obtained. By repeating this process a number of points on the required path can be obtained sufficiently close together to draw a curve through them. Figures 220, 221, and 222

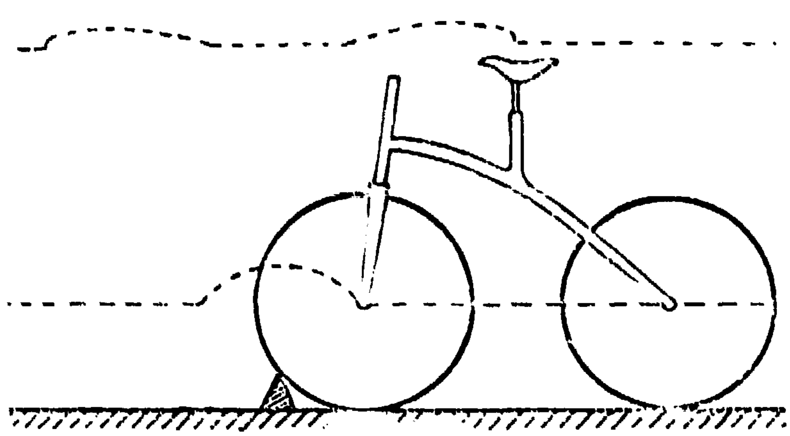


FIG. 222.

respectively show the curves described by a point a short distance above the saddle of an 'Ordinary,' of a 'Rear-driving Safety' with wheels 28 in. and 30 in. diameter, and of a 'Bantam' with both wheels 24 in. diameter, the point being midway between the wheel centres. A number of such curves are given and exhaustively discussed in R. P. Scott's 'Cycling Art, Energy, and Loco-

motion,' though it should be noticed that the curved portions of the saddle paths, due to the front and rear wheels passing over the obstruction, are shown placed in wrong positions.

191. **Influence of Size of Wheel.**—In figure 220 it will be noticed that the total heights of the curved portions of the paths of the wheel centres above the straight portions are the same, whatever be the diameter of the wheel; but the greater the diameter of the rolling wheel, the greater is the horizontal distance moved over by the wheel centre in passing over the stone. Thus with a large wheel the stone is mounted and passed over more gradually, and therefore with less shock, than with a small wheel. Therefore, other things being the same, large wheels are better than small for riding over loose stones lying on a good flat road.

192. **Influence of Saddle Position.**—The motion of the saddle may be conveniently resolved into vertical and horizontal components. In riding along a level road the vertical motion is zero and the horizontal motion uniform. When the front wheel meets an obstacle the motion of the frame may be expressed as a motion of translation equal to that of the rear wheel centre, Q, together with a motion of rotation of the frame about Q as centre. Let ω be the angular speed of this rotation at any instant. The linear motions of P and R relative to Q will be in directions at right angles to $Q P$ and $Q R$ respectively, and their speeds will be $\omega \times Q P$ and $\omega \times Q R$ respectively; the lines $Q P$ and $Q R$ (fig. 223) may therefore represent the magnitudes of the velocities, the directions being at right angles to these lines. Through Q draw a horizontal line, and to it draw perpendiculars $P p$ and $R r$. Then $Q p$ and $Q r$ will represent the vertical components of the motions of P and Q respectively, $P p$ and $R r$ the horizontal components.

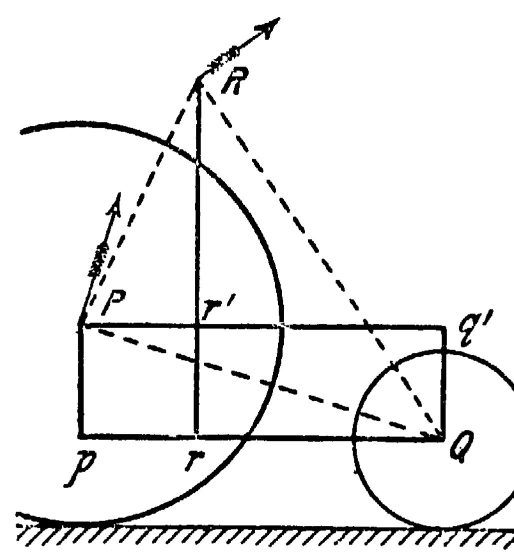


FIG. 223.

In the same way, if the front wheel be moving along the level,

and the back wheel be passing over an obstacle, by drawing perpendiculars $R r^1$ and $Q q^1$ to a horizontal line through P, it can be shown that $P q^1$ and $P r^1$ represent the vertical components of the motions of Q and R respectively relative to P, $Q q^1$ and $R r^1$ the horizontal components.

Therefore, in a bicycle with equal wheels, the vertical 'jolting' communicated to the saddle by one of the wheels passing over an obstacle is proportional to the horizontal distance of the saddle from the centre of the other wheel, the horizontal 'pitching' to the vertical distance from the centre of the other wheel. With wheels of different sizes the average angular speeds ω are inversely proportional to the chords $A B$ and $a b$ (fig. 220); this ratio must be compounded with that mentioned above.

If the saddle of a tricycle be vertically over the centre of the wheel-base triangle, its vertical motion will be one-third that of one of the wheels passing over a stone. In the 'Rudge' quadricycle the vertical motion would be one-fourth, with similar conditions as to position of saddle.

From the above discussion it is readily seen that the most comfortable position for the saddle, as regards riding over rough roads, is midway between the wheel centres, the vertical motion of the saddle being then half that of a wheel going over a stone. In a tandem, with one seat outside the wheel centres, the vertical jolting of this seat is greater than that of the nearer wheel. Again, as regards horizontal pitching, the high bicycle compares unfavourably with the low; the rider on the top seat of the 'Eiffel' bicycle would have to hold on hard to avoid being pitched clean out of his seat while riding fast over a rough road. A long wheel-base is a decided advantage as regards horizontal pitching in riding over stones. The angular speed ω of the frame in mounting over a stone is, other conditions remaining the same, inversely proportional to the length of the wheel-base. Therefore, the pitching is also inversely proportional to the length of the wheel-base.

A curious point may be noticed in the case of the 'Ordinary.' From the saddle path shown (fig. 220) it will be seen that when the rear wheel, after surmounting the obstacle, is descending again to the level, the saddle actually moves backwards. This

can only happen at slow speeds; at higher speeds the rear wheel actually leaves the stone before touching the ground, and the backward kink in the saddle path may be eliminated.

193. **Motion over Uneven Road.**—If the surface of the road be undulating, but free from loose stones, the paths of the wheel centres, *P* and *Q*, will be curves parallel to that of the road surface, and the path of any point rigidly fixed to the frame can be found by the same method. In a very bad case, the undulations being very close together (fig. 224), it may happen that the radius of curvature of one of the holes is less than the radius of a large bicycle wheel.

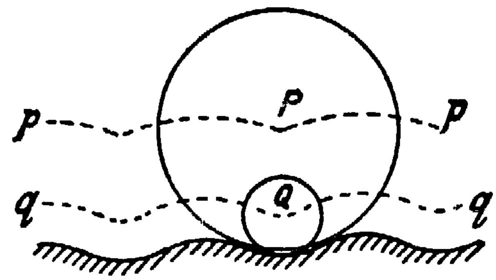


FIG. 224.

In this case the path, *p p*, of the large wheel will have abrupt angles, while that of the smaller wheel, *q q*, may be continuous, the large wheel being actually worse than the small one.

194. **Loss of Energy.**—If the motion of a wheel over an obstacle took place very slowly, there would theoretically be no loss of energy in passing over it, since the work done in raising

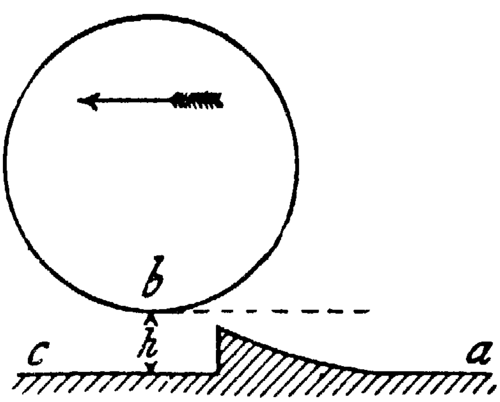


FIG. 225.

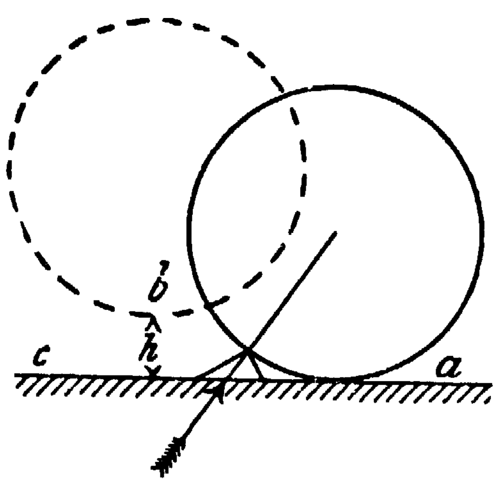


FIG. 226.

the weight would be restored as the weight descended; but at appreciable speeds the loss of energy by impact and shock may be considerable. Let a wheel moving in the direction of the arrow (fig. 225) pass over an obstacle of such a form that the wheel rises without sudden jerk or shock to a height *h*, the speed being so great that at its highest point the wheel is clear both of

the obstacle and the ground. If W be the weight (including that of the wheel) resting on the axle, the energy lost will be $W h$, since the kinetic energy in position b is this amount less than that in position a. The energy due to the fall from b to c is wasted in shock, there being no means of obtaining a forward effort from the work done during the descent.

If the wheel strike the obstacle suddenly (fig. 226) and then rises to the height h, clear of the ground and obstacle, the energy lost may be greater than $W h$, the amount depending on the nature of the surface of the wheel tyre and the obstacle struck.

If the horizontal speed of the wheel be such that it does not leave contact with the obstacle in passing over it, the nature of the losses of energy can be shown as follows:

The centre of the wheel at the instant of coming into contact with the stone, S (fig. 227), is moving with velocity v in a horizontal direction. This can be resolved into a velocity v_1 in the direction $c_1 S$, joining the wheel centre to the stone, and a velocity v_2 at right angles to this direction. The velocity, v_1, is the velocity of impact of the wheel on the stone S, and the energy due to this velocity may be entirely lost.

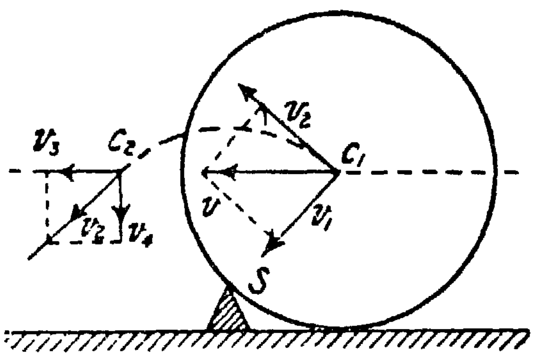


FIG. 227.

If e be the index of elasticity, the velocity of rebound is $e v_1$, and with suitable elastic tyres the energy due to this velocity may be saved. The loss of energy due to the impact on the stone will be at least (sec. 69)

$$(1 - e^2) \frac{m v_1^2}{2 g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and may be as great as

$$\frac{m v_1^2}{2 g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where m is the weight of the portion of the machine rigidly connected to the wheel tyre.

The motion of the wheel continuing, the wheel centre mounts over the stone, describing a circle, $c_1 c_2$, with centre, S, and the

tyre will again touch the ground on a point in front of the stone. If the speed of the machine be uniform, the velocity of the wheel centre as the wheel again just touches the ground may be equal in magnitude to v_2, the same as immediately after impact on the stone. This velocity, v_2 (fig. 227), can be resolved into horizontal and vertical components, v_3 and v_4. v_4 is the velocity of impact on the ground, and the energy due to it is either partially or entirely lost, and the final velocity of the wheel centre is v_3.

The assumption made above, that the speeds of the wheel centre, C, when in positions c_1 and c_2 are equal, is equivalent to assuming that the reactions of the stone on the wheel in any position before passing the vertical line through the stone is exactly equal to the reaction when at an equal distance past the stone ; or, briefly, the reactions as the wheel rolls on and off the stone are equal. With a hard unyielding tyre this is not even approximately true, except at very low speeds, consequently the positive forward effort exerted on the wheel as it rolls off the stone is less than the backward effort exerted as it rolls on, and the speed is seriously diminished. With a tyre that can adapt itself *instantaneously* to the inequalities of the road, the reactions during rolling on and off a stone are equal, and there is no loss of energy. The pneumatic tyre is the closest approximation to such an ideal tyre, while rubber is much better than iron.

If the road surface be undulating, the undulations being so long that the path of the wheel centre is a curve with no sudden discontinuities, there may be no loss of energy due to the undulations. If the undulations, however, be so short, and the speed of the machine so great, that the wheel after ascending an undulation actually leaves contact with the ground, there will be a loss of energy due to the impact on reaching the ground.

CHAPTER XX

RESISTANCE OF CYCLES

195. **Expenditure of Power.**—The energy a cyclist generates while riding along a level road is expended in overcoming the various resistances to motion. These may be classed as follows: (1) Friction of bearings and gearing of the machine. (2) Rolling resistance of the wheels on the ground. (3) Resistance due to loss of energy by vibration. (4) Resistance of the air. The power expended in overcoming these resistances is the power actually communicated to the machine, and may be called the *brake* power of the rider. The power actually generated in the living heat-motor (the rider's body) may be called the *indicated* power; the difference between the *indicated* and the brake powers will be the power spent in overcoming the frictional resistance of the motor—*i.e.* the friction of the rider's joints, muscles, and ligaments. At very high pedal speeds the brake power is small compared with the indicated; in fact, by supporting the bicycle conveniently, taking off the chain, and pedalling as fast as he can, a rider may possibly develop more indicated power than when racing on a track, though the brake power is practically zero. The gearing of the bicycle, therefore, must not be made too low, or the greater part of the rider's energy will be spent in heating himself. The estimation of the work so wasted lies in the domain of the physiologist rather than in that of the engineer; we proceed, therefore, to the consideration of the brake power and its expenditure.

196. **Resistance of Mechanism.**—The frictional resistance of the bearings is very small compared with the other resistances to be overcome; the resistance due to friction of the bearings of a bicycle moving on a smooth track is practically the same at all

speeds. Professor Rankin estimated this at $\frac{1}{1000}$ part of the weight of the rider, but exact experiments are wanting.

The frictional resistance of the chain possibly varies with the pull on it, and as, other things being equal, the pull of the chain increases with the speed, the resistance will also vary with the speed. However, in comparison with the resistance due to rolling and with the air resistance, that of the chain is small, and may be included in the internal resistance of the machine, which we may say is approximately constant at all speeds.

197. **Rolling Resistance.**—The resistance to rolling is, according to the experiments of Morin, composed of two terms, one constant, the other proportional to the speed. With a pneumatic tyre on a smooth road the second term is negligible in comparison with the first, according to M. Bourlet. The rolling resistance is inversely proportional to the diameter of the wheel.

In 'Traité des Bicycles et Bicyclettes,' C. Bourlet says that the rolling resistance with pneumatic tyres is small, independent of the speed, and on a dry road it varies from

$$\cdot 005\ W \text{ to } \cdot 01\ W \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

while on a racing track the probable value for the resistance is $\cdot 004\ W$, W being the total weight of machine and rider.

The resistance of a solid rubber tyre varies with the speed, and may possibly be expressible by a formula of the form

$$R = A + B\,v, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

A and B being constants.

The power P required to overcome the rolling resistance $\cdot 005\ W$ at the speed v is

$$P = \cdot 005\ W v \text{ units} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If W be expressed in lbs. and v in miles per hour,

$$P = \cdot 44\ W v \text{ foot-lbs. per min.} \quad . \quad . \quad (4)$$

198. **Loss of Energy by Vibration.**—One of the great advantages of a pneumatic tyre is that little or no vibration is communicated to the machine and rider. On a smooth road or track with pneumatic tyres the loss due to vibration is probably negligible; but on a rough road it may be very large, and is

possibly proportional to the speed. With solid tyres, a considerable amount of energy is lost in vibration. Bourlet's experiments on the road show that the work wasted in vibration is about one-sixth of the total.

The use of a pneumatic tyre enables the tremulous vibration to be almost eliminated, no vibration being communicated to any part of the machine. For riding over very rough roads the introduction of springs into the wheel or frame may still further diminish vibration. The anti-vibrators should be placed so that they protect as great a portion of the machine from vibration as possible. In this respect a spring wheel should be better than a spring frame, and a spring frame, in turn, better than a spring saddle. The machine, as a whole, should be made sufficiently strong and rigid that none of its parts yield under the stresses to which they are subjected. Of course, when a spring yields and again extends, a certain amount of energy is lost; it thus becomes a question as to when springs are advantageous or otherwise. Probably the rougher the road, the more can springs be used with advantage in the wheels, frame, and saddle; whereas, on a smooth racing track, their continual motion would simply provide means of wasting a rider's energy.

199. **Resistance of the Air.**—M. Bourlet discusses the air resistance of a rider and machine, and concludes that it may be represented by a formula

$$R = k\,S\,v^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (5)$$

R being the air resistance, S the area of the surface exposed, v the speed, and k a constant. If the resistance be measured in kilogrammes, the area in square metres, and the speed in metres per second, $k = \cdot 06$. The area of surface exposed will depend on the size of the rider and his attitude on the bicycle. A mean value for S is $\cdot 5$ square metre; then

$$R = \cdot 03\,v^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

If the resistance be measured in lbs., and the speed V in miles per hour,

$$R = \cdot 013\,V^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (7)$$

The power required to overcome this resistance is

$$1 \cdot 144 \ V^3 \text{ foot-lbs. per minute} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Table X. gives the air resistances and the corresponding powers at different speeds calculated from these formula.

TABLE X.—AIR RESISTANCE TO 'SAFETY' BICYCLE AND RIDER.

Speed	Resistance	Power	Speed	Resistance	Power
Miles per hour	lbs.	Foot-lbs. per min.	Miles per hour	lbs.	Foot-lbs. per min.
5	·32	143	18	4·21	6,672
6	·47	247	19	4·69	7,846
7	·64	392	20	5·20	9,152
8	·83	586	21	5·73	10,600
9	1·05	834	22	6·29	12,180
10	1·30	1,144	23	6·88	13,920
11	1·57	1,522	24	7·49	15,820
12	1·87	1,977	25	8·12	17,870
13	2·20	2,513	26	8·79	20,100
14	2·55	3,139	27	9·48	22,520
15	2·92	3,861	28	10·19	25,110
16	3·33	4,685	29	10·93	27,900
17	3·76	5,620	30	11·70	30,890

If the wind be blowing exactly with or against the cyclist, his speed *relative to the air* must be used in the above formula. Thus, if the wind be blowing at the rate of 10 miles per hour, and the rider be moving at the rate of 20 miles per hour, while going *against* the wind, the air resistance is that due to a speed of 30 miles per hour, while going with the wind there is still a resistance due to a speed of $20 - 10 = 10$ miles per hour.

If v be the speed of the cyclist, V that of the wind, while riding against the wind the relative speed is $(v + V)$. If the cyclist rides at a high speed, a very slight breeze against him may increase the air resistance considerably. Whilst riding with the wind the relative speed is $(v - V)$. In this case, if the speed of the wind be greater than that of the cyclist, there will be no resistance, but, on the contrary, assistance will be afforded by the wind. If the speed of the wind be less than that of the cyclist, there will be air resistance due to the speed $(v - V)$.

The power required to overcome air resistance in driving at v miles per hour against a wind blowing V miles an hour is

$$P = 1{\cdot}144\, v\, (v + V)^2 \text{ foot-lbs. per minute} \quad . \quad . \quad (9)$$

that required in going with the wind,

$$P = 1{\cdot}144\, v\, (v - V)^2 \text{ foot-lbs. per minute} \quad . \quad . \quad (10)$$

This equation gives also the power expended in overcoming air resistance by a rider behind pace-makers ; the principal beneficial effect of pace-makers being to create a current of wind of speed V assisting the rider.

With a side wind blowing, the air resistance is greater than that due to the relative speed. In moving through still air, or against a head wind, the cyclist drags with him a certain quantity of air. A side wind has the effect of changing very rapidly the actual particles dragged by the cyclist, so that in a given period of time the mass of air which has to be impressed with the rider's speed is greater than with a head wind of the same speed. Hence an increased resistance is experienced by the rider.

A consideration of the figures in Table X. will show that bicycle record-breaking depends more on pace-making arrangements than on any other single factor. For example, to ride unpaced at twenty-seven miles an hour requires the expenditure of more than two-thirds of a horse-power to overcome only the air resistance. Though an average speed of $27\frac{1}{2}$ miles per hour was kept up by Mr. R. Palmer and by Mr. F. D. Frost in the Bath Road Club 100-miles race, 1896, it is most improbable that they worked at anything like this rate during the whole period, the difference being due to the decrease in the air resistance caused by the pace-makers in front.

200. **Total Resistance.**—Summing up, the total resistance of the bicycle can be expressed by the formula

$$R = A + B\,v + C\,v^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and the power required to drive it by

$$P = A\,v + B\,v^2 + C\,v^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

A, B, and C being co-efficients depending on the nature of the mechanism and the condition of the road, but which are constant for the same machine on the same road at different speeds.

Figure 228 shows graphically the variation of the power required to propel a cycle as the speed increases. The speeds are set off as abscissæ. For any speed, $O\ S$, the power required to overcome the frictional resistance of the mechanism is set off as an ordinate $S\ M$; the power required to overcome rolling resistance is $M\ T$ (W being taken at 180 lbs.); the power required

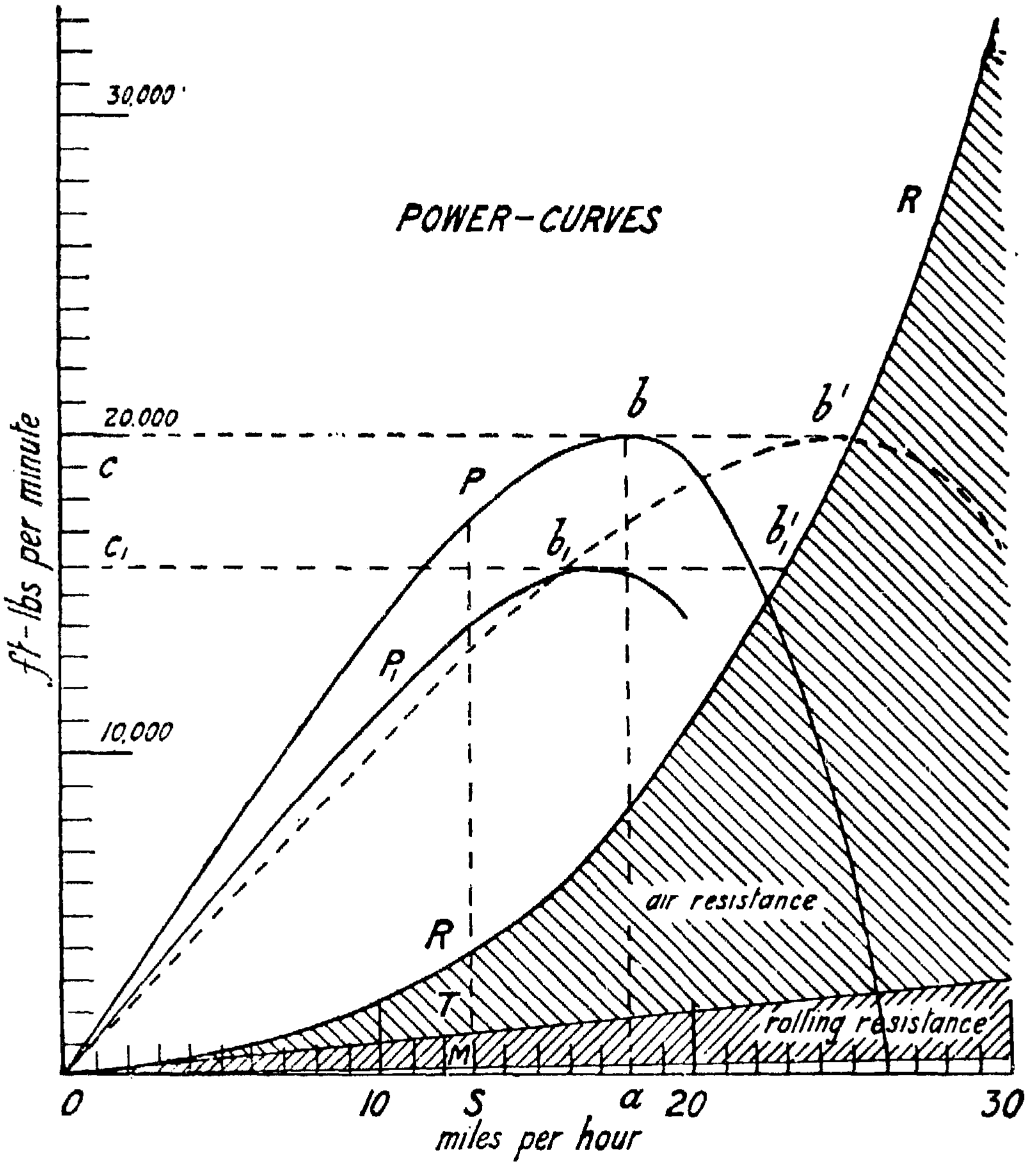


FIG. 228.

to overcome air resistance is $T\ R$; and the total power required is the ordinate $S\ R$. The curve M can be lowered by improvements in the mechanism, the curve T by improvements in the tyres and track-surface, and the curve R by improvements in pace-making.

Experiments on the total resistance of a cycle can be carried

out in two ways. Firstly, by towing the machine and rider along a level road by means of another machine, the pull on the tow-line being read off from a spring-balance. Secondly, by letting the machine and rider run down a hill, the gradient of which is known, until a uniform speed is attained; the ratio of the resistance at the speed attained to the total weight of machine and rider is the sine of the angle of inclination of the road. The second method is not convenient for a series of experiments at different speeds, since a number of hills of different gradients are required; but since no extra assistance is required, a rider may use it when unable to use the first method.

Table XI., taken from 'Engineering,' January 10, 1896, giving results of experiments by Mr. H. M. Ravenshaw, serves to show the variation of the resistance according to the state of the road.

TABLE XI.—RESISTANCE OF CYCLES ON COMMON ROADS.

Machine	Road	Total weight. Lbs.	Pounds per ton	Miles per hour
Tandem Tricycles, Pneumatic Tyres	Flint . . .	120	37	4
	,, . . .	290	31	4
	,, . . .	290	31	10·4
	,, . . .	290	31	7
	,, . . .	440	35	4
	,, . . .	440	35	8·3
	Asphalte pavement	290	31	4
	,, ,,	440	30	4
	,, ,,	440	30	6
	Heavy mud . .	290	73	4
	Wet mud . .	290	65	12
Tandem Bicycles, Pneumatic Tyres	Flint . . .	200	33	5
	,, . . .	370	30	5
	Heavy mud . .	200	95	5
	,, ,, . .	370	78	5
	Flag pavement .	200	33	5
Single Tricycles, Solid Tyres . .	Flint . . .	220	60	4
	,, . . .	220	60	8
	Flag pavement .	220	60	5
	Heavy mud . .	200	146	4

CHAPTER XXI

GEARING IN GENERAL

201. **A Machine** is a collection of bodies designed to transmit and modify motion and force. The moving parts of a machine are so connected, that a change in the position of one piece involves, in general, a certain definite change in the position of the others. A bicycle or tricycle is a machine in which work done by the rider's muscles is utilised in changing the position of the machine and rider. Coming to narrower limits, we may say a cycle is a machine by which the oscillatory movement of the rider's legs is converted into motion of rotation of a wheel or wheels rolling along the ground, on which is mounted a frame carrying the rider. Still more narrowly, we may consider a cycle as a mechanism for converting the motion of the pedals, which may be either oscillatory or circular, into motion of rotation of the driving wheel.

202. **Higher and Lower Pairs.**—Each part of a machine must be in contact with at least one other part; two parts of a mechanism in contact and which may have relative motion forming a *pair*. If the two parts have contact over a surface, as is necessary when heavy pressures are transmitted, the pair is said to be *lower*. From this definition there can only be three kinds of lower pairs—turning pairs, sliding pairs, and screw pairs; as in a shaft and its journal, a cylinder and piston, a bolt and its nut, respectively. If the elements of a pair do not have contact over a surface, or if one of the elements is not rigid, the pair is said to be *higher*, the relative motion of the pair being, as a rule, much more complex than that of lower pairs. A pair of toothed-wheels in contact, a flexible band and drum, a ball and its bearing-case, are examples of higher pairs.

Link or Connector.—Two elements of consecutive pairs may be connected together by a *link*. An assemblage of pairs connected by links constitute a *kinematic chain*, or a *mechanism*, or a *gear*. The simplest kinematic chain contains four pairs connected by four links; it is therefore called a four-link mechanism. If one link be fixed, a motion given to a second link will produce a determinable motion of the two remaining links. Three pairs united by three links constitute a rigid triangle, while a five-link chain requires further constraint for movement of a definite character to be produced. The four-link kinematic chain is the basis of probably 99 per cent. of all linkwork mechanisms.

203. **Classification of Gearing.**—Professor Rankine defines an elementary combination in mechanism as a pair of primary moving pieces so connected that one transmits motion to the other; that whose motion is the cause is called the driver, the other the follower. The connection between the driver and follower may be:

(1) By rolling contact of their surfaces, as in toothless wheels.

(2) By sliding contact of their surfaces, as in toothed-wheels and cams, &c.

(3) By flexible bands, such as belts, cords, and gearing chains.

(4) By linkwork, such as connecting-rods, &c.

(5) By reduplication of cords, as in the case of ropes and pulleys.

(6) By an intervening fluid.

The driving gear of cycles has been made from classes (2), (3), and (4), each of which will form the subject of a separate chapter. An example of (1) is found in the 'Rotherham' cyclometer, the wheel of which is driven by rolling contact from the tyre of the front wheel. The pump of a pneumatic tyre is an example of (6). We cannot recollect an example in cycle construction corresponding to (5), though it would be easy to design one to work in connection with a pedal clutch gear, such as the 'Merlin.'

204. **Efficiency of a Machine.**—If the pairs of a mechanism could perform their relative motion without friction, the work done by the prime mover at the driving end of the machine would be transmitted intact to the driven end; in other words, the work got out of the machine would be equal to that put into

it. But however skilfully the parts be designed to reduce friction to the lowest possible amount, there is always *some* frictional resistance which consumes energy, so that the work got out of the machine is less than that put into it, by the amount of work spent in overcoming the frictional resistance of the pairs.

The ratio of the work transmitted by the machine to that supplied to it is called the *efficiency of the machine.* The efficiency of a machine will be higher according as the number of its pairs is small; an increase in the number of pairs increases the opportunities for work to be wasted away. Thus, in general, the simpler the mechanism used, the better will be the results obtained.

It seems perhaps unnecessary to say that no advantage can be derived from mere complexity of mechanism, but the number of driving gears for cycles that are being patented shows either that the perpetual motion inventor has plenty of vitality, or that the technical common sense of a large number of cycle purchasers is not of a very high standard.

205. **Power.**—We have already seen that the work done by an agent is the product of the applied force, into the distance through which the point of application of the force is moved in the direction of the applied force. The *power* of an agent is equal to the *rate* of doing work—that is, power may be defined as the *work done per unit of time.* If E be the work done in t seconds, and P the power of the agent, then

$$P = \frac{E}{t}.$$

But E is equal to $F s$, where F is the force acting and s the distance moved; therefore

$$P = \frac{F s}{t}.$$

But $\frac{s}{t}$ is equal to the speed; therefore

$$P = F v \qquad \text{. . . . . (1)}$$

That is, the power of the agent is equal to the product of the acting force and the speed of its point of application. The same

principle is expressed in the maxim, 'What is gained in power is lost in speed'; the word 'power' in this maxim having the meaning we have associated with 'force' throughout this book.

In a frictionless machine the power is transmitted without loss. The above equation shows that any given horse-power may be transmitted by any force F, however small, provided the speed v can be made sufficiently great. On the other hand, if the speed of transmission be very small, a very large force, F, may correspond to a very small transmission of power. An example of the former case occurs in transmitting power to great distances by means of wire rope. Here the speed of the rope is made as large as it is found practicable to run the pulleys, so that a rope of comparative small diameter may transmit a considerable amount of power. An example of the latter case occurs in a hydraulic forging press, where the pressure exerted on the ram is, in many cases, 10,000 tons; but the speed of the ram being small - only a few inches per minute - the horse-power required to work such a press may be comparatively small.

These principles are of direct application to the gearing of cycles.

Example I. Suppose two rear-driving bicycles each to have 28-inch driving-wheels geared to 56 inches; let the bicycles be equal in every respect, except that in one the numbers of teeth in the wheels on the crank-axle and hub are 16 and 8 respectively, while in the other the numbers are 18 and 9 respectively. When going along the same gradient at the same speed, the speeds of the chain relative to the machine are in the ratio of 8 to 9; consequently, the pulls on the chain will be in the ratio 9 to 8, that on the chain of the bicycle having the smaller wheels being the greater.

Example II.- Let two bicycles be the same in every respect, except that in one the cranks are 6 inches long, in the other 7 inches. When running along the same road at the same speed, the work done in overcoming the resistance will be the same in the two cases, and, therefore, the work done by the pressure of the feet on the pedals is the same in both cases. But the pedals' speeds are in the ratio of 6 to 7, therefore the average pressures

to be applied to the pedals are in the ratio 7 to 6, the shorter crank requiring the greater pressure.

Example III.—Suppose two Safety bicycles to be equal in every respect, except that one is geared to 56 inches, the other to 63 inches. With equal riders, running along the same road at the same speed, the work done in both cases will be equal. But the distances moved over during one revolution of the crank are in the ratio of 56 to 63, that is, 8 to 9. The numbers of revolutions required to move over a given distance will therefore be in the ratio of the reciprocals of the distance—that is, 9 to 8. Consequently, the average pressures to be applied to the pedals in the two cases will be in the ratio of 8 to 9, the bicycle with the low gear requiring the smaller pressure on the pedals

The whole question of gear for a bicycle thus resolves itself into a question of what will suit best the convenience of the rider. Assuming that the maximum power of two riders is exactly the same, one may be able to develop his maximum power by a comparatively light pressure on the pedals and a high speed of revolution of the cranks, the other may develop his maximum power with a heavier pressure and a smaller speed of revolution of the crank-axle. The former would therefore do his best work on a lower geared machine than the latter. The question of length of crank depends also on the same general principles, different riders being able to develop their maximum powers on different lengths of crank.

The maximum power a rider can develop by pedalling a crank-axle is probably at low speeds proportional to the speed of driving ; at higher speeds the power does not increase so rapidly as the speed, and soon reaches an absolute maximum ; at still higher speeds the rapidity of pedalling is too great, and the power actually communicated to the crank-axle rapidly falls to zero. These variations of the power with the speed are graphically represented by the curves P and P_1 (fig. 228), P_1 being for longer sustained effort than P ; a certain speed of the crank-axle corresponding to a definite speed of the cycle on the path, so long as the gearing remains unaltered. The height of the ordinates will depend on the duration of the ride, and the maximum power $a\,b$ for an effort of short duration may be developed at a less axle

speed than the maximum $a_1 b_1$ for a longer effort. By increasing the amount of gearing-up, the abscissæ of the curve would be all proportionately increased, while the ordinates remain as before. The best gearing-up possible for the rider will be such that the power curve of the machine intersects the rider's power curve at the highest point of the latter. From b, the highest point of the rider's power curve with a certain gearing-up, draw $b\,b^1$ to intersect at b^1 the power curve R of the machine, then the rider will develop the greatest speed $c\,b^1$ on the machine if the gearing-up be increased in the ratio of $c\,b$ to $c\,b^1$. If, as seems to the author most probable, the ratio $\frac{c\,b^1}{c\,b}$ for the shorter effort is greater than the ratio $\frac{c_1 b^1{}_1}{c_1 b_1}$ for the longer effort, the gearing-up should be greater for the former than for the latter. That is, to attain in all races his highest possible speed, the shorter the distance the higher should be the gear used by the rider.

Very little is known as to the maximum power that can be developed by a cyclist, no accurate experiments, to the author's knowledge, having been made. Rankine gives 4,350 foot-lbs. per minute as the average power of a man working eight hours raising his own weight up a staircase or ladder, and 17,200 foot-lbs. per minute in turning a winch for two minutes. Possibly racing cyclists of the front rank develop for short periods two-thirds of a horse-power—*i.e.* 22,000 foot-lbs. per minute. If this estimate and that of the air resistance (sec. 199) be correct, from figure 228 it is evident that a speed of 28 miles per hour could not be attained on a single bicycle, in still air, without pace-makers, even though the mechanism and the tyres were theoretically perfect. It should be noted that the conventional horse power, 33,000 foot-lbs. per minute, introduced by Watt, and employed by engineers as the unit of power, is considerably in excess of the average power of a draught horse.

206. **Variable-speed Gear.**—The maximum power of any rider is exerted at a particular speed of pedal and with a particular length of crank. The best results on all kinds and conditions of roads would probably be attained if the pedal could always be kept moving at this particular speed whatever the resistance; the

gearing would then have to vary the distance travelled over per stroke of pedal, until equilibrium between the effort and resistance was established. An ideal variable gear would be one which could be altered continuously and automatically, so that when going uphill a low gear was in operation, and when going down hill a high gear. A number of two-speed gears have been used with success, and are described in chapter xxvii., but no continuously varying gear has been used for a cycle driving gear, though such a combination is well known in other branches of applied mechanics.

207. **Perpetual Motion.**—Many inventors and schemers do not appreciate the importance of the principle of 'what is gained in force, or effort, is lost in speed.' Since for a given power the effort or force can be increased indefinitely by suitable gearing, and likewise the speed, they appear to reason that by a suitably devised mechanism it may be possible to increase both together, and thus get more power from the machine than is put into it. A crank of variable length, the leverage being greater on the down than on the up-stroke, is a favourite device. The Simpson lever-chain is another device having the same object in view. The angular speeds of the crank-axle and back hub are inversely proportional to their numbers of teeth; with an ordinary chain the distances of the lines of action from the centres are directly proportional to these numbers. By driving the back hub chain-wheel from pins on the chain links at a greater distance from the wheel centre, it was claimed that an increased leverage was obtained, and that the lever-chain was therefore greatly superior to the ordinary. It is possible, by using an algebraic fallacy which may easily escape the notice of anyone not sufficiently skilled in mathematics, to prove that $2 \times 2 = 5$; but though the human understanding may be deceived by the mechanical and algebraic paradoxes, in neither case are the laws of Nature altered or suspended. When once the doctrine of the 'conservation of energy' is thoroughly appreciated, plausible mechanical devices for *creating* energy will receive no more attention than they deserve.

208. **Downward Pressure.**—In all pedomotive cycles the *general* direction of the pressure exerted by the rider on the pedals

is vertically downwards. If P be the average vertical pressure and d the vertical distance between the highest and lowest points of the pedal's path, the work done by the rider per stroke of pedal is Pd. This is quite independent of the form of the pedal path.

209. **Cranks and Levers.**—If the pedals are fixed to the ends of cranks revolving uniformly, the vertical component of the pedal's motion will be a simple harmonic motion, and, neglecting ankle action, the motion of the rider's knee will be approximately simple harmonic motion along a circular arc.

When the crank is vertical, its direction coincides with that of the vertical pressure, and consequently no pressure, however great, will tend to drive the crank in either direction. The crank is then said to be on a 'dead-centre.' In steam-engines, and mechanisms in which the crank is employed to convert oscillating into circular motion, a fly-wheel is used to carry the crank over the dead-centre. In cycles, when speed has been got up, the whole mass of the machine and rider tends to continue the motion, and thus acts as a fly-wheel carrying the crank over the dead-centre, so that in riding at moderate or high speeds the existence of the dead-centre is hardly suspected. In riding at a very slow speed, however, the existence of the dead-centre is more manifest. If two cranks are placed at right angles to each other on the same shaft, while one is on the dead-centre the other is in the best position for exerting the downward effort, and there is no tendency of the shaft to stop.

In the above discussion we have assumed that the connecting-rod which drives the crank can only transmit a simple thrust or pull; if, in addition to this, the connecting-rod can transmit a transverse effort there may be no dead-centre. In turning the handle of a winch by hand, the arm acts as a connecting-rod which can transmit, thrust, pull, and transverse effort, so that no dead-centre exists. In Fleming & Ferguson's marine-engine two cylinders are connected by piston-rods and intermediate links to two corners of a triangular connecting-rod, the third corner of which is at the crank; with this arrangement there is no dead-centre, the single crank and triangular connecting-rod being in this respect equivalent to two cranks at right angles.

The existence of the dead-centre is supposed by some to be

a disadvantage inherent to the crank, but the efficiency of the mechanism is not in any way directly affected by it.

210. **Variable Leverage Cranks.**—One favourite notion of those inventors who have no clear and exact ideas of mechanical principles, is to have a crank of variable length arranged so that the leverage may be great during the down-stroke of the pedal and small during the up-stroke ; their idea evidently being to obtain all the mechanical advantages of a long crank, and yet only make the foot travel through a distance corresponding to a short crank. We have shown above that, presuming the pressure is vertical, the work done per stroke of pedal depends only on the pressure applied, and the vertical distance between the highest and lowest points of the pedal path ; the distance of the pedal from the centre of the crank-spindle having no *direct* influence whatever. The pedal path in most of the variable crank gears that have appeared from time to time is simply an epicycloidal curve which does not differ very much in shape from a circle, but which is placed nearer the front of the machine than an equal circle concentric with the crank-axle. Thus, the gear only accomplishes in a clumsy manner what could be done by a simple crank, having its axle placed a little further forward than that of the variable crank.

Let O (fig. 229) be the centre of a variable crank, and $c\,d$ the pedal path during the up-stroke. Let the length of the crank become greater, the path of the pedal during this extension being $d\,a$, and let the arc $a\,b$ be the pedal path during the down-stroke. The crank will then shorten, $b\,c$ being the pedal path. If the pressure be vertically downward, work will be done only while the pedal moves from a to b, and the angle of driving will be the small angle $a\,o\,b$. Thus while with a variable crank a greater turning effort may be exerted than with a fixed crank, the arc of action is correspondingly less.

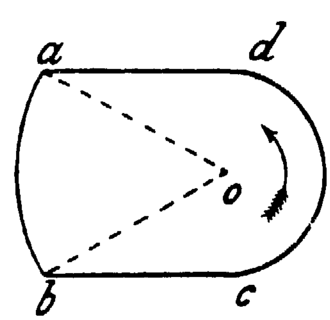


FIG. 229.

211. **Speed of Knee-joint during Pedalling.**—Regarding that part of the leg between the knee and the foot as a connecting-rod, that between the knee and the hip-joint as a lever vibrating about a fixed centre, the speed of the knee corresponding to a uniform speed of the pedal can easily be determined by the method of

section 33. Figure 23 is a polar curve showing the varying speed of the knee for different positions of the crank. From this curve it will be seen that on the down-stroke the maximum speed is attained when the crank is nearly horizontal, but on the up-stroke the maximum speed is not attained till the crank is nearly 45° above the horizontal. The speed then rapidly diminishes, and is nearly zero when the crank is vertical. The shorter the crank, in comparison with the rider's leg, the more closely does the motion of the knee approximate to simple harmonic motion ; with simple harmonic motion the polar curve is two circles.

In any gear in which a crank connected to the driving-wheel is used, the speed of the knee-joint will vary approximately as above described—*i.e.* it will gradually come to rest as it approaches its highest and lowest positions, then gradually increase in speed until a maximum is attained.

212. **Pedal-clutch Mechanism.**—Instead of cranks, clutch gears have been used for the driving mechanism. In these a cylindrical drum is placed at each side of the axle and runs freely on it. A long strap, with one end firmly fixed to the drum, is coiled once or twice round it, the other end is fastened to the pedal lever. When the pedal is depressed, the drum is automatically clutched rigidly to the shaft ; when the pressure is removed from the pedal, the pedal lever is raised by a spring and the drum released from the axle. One of the most successful clutch gears was that used on the 'Merlin' bicycles (fig. 176) and tricycles made by the Brixton Cycle Company.

The general advantage which a clutch gear was supposed to have as compared with a crank was that any length of stroke could be taken from a pat of an inch up to the full throw of the gear. However, even supposing that the clutches which lock the drums to the axle and the springs which lift the pedal levers are perfect in action, the gear has the serious defect that the down-stroke of the pedal begins quite suddenly and is performed at a constant speed ; thus the legs must have a considerable speed imparted suddenly to them. At moderate and high speeds this is a decided disadvantage as against the gradual motion required for the crank-geared cycle. There is the further serious practical disadvantage that no clutch that has been hitherto designed is

perfectly instantaneous in its action of engaging and disengaging. When a clutch is used for continual driving, as in the clutch driving gears of some of the early tricycles, and where no great importance need be attached to the delay of a second or two in the action of the clutch gear, the case is quite different. Mr. Scott, in 'Cycling Art, Energy, and Locomotion,' has put the comparison between the crank gear and clutch gear for pedals in a nutshell thus: "In the crank-clutch cycle, as in other uses, the immediate solid grip is a matter of very little concern; if a half turn of the parts takes place before clutching, it does very little harm, since it is so small a fraction of the entire number of revolutions to be made before the grip is released. But if a grip is to be taken at every down-stroke of the foot, as in a lever-clutch cycle, the least slip or lost motion is fatal."

These two objections are so weighty, that in spite of the immense advantage of providing a simple variable gear, pedal-clutch gears have never been much used.

213. **Diagrams of Crank Effort.**—Though the pressure on the pedal may be constant during the down-stroke, the effort tending to turn the crank will vary with the varying crank position. The actual pressure on the pedal may be resolved into two components, parallel and at right angles to the crank; the former, the radial component, merely causes pressure on the bearing, and, since no motion takes place in its direction, no work is done by it; the latter, the tangential component, constitutes the active effort tending to turn the crank. If $O\,C$ (fig. 230) be the crank in any position, and P the total pressure on the pedal, the radial and tangential components, R and T, are equal to the projections of P respectively parallel to, and at right angles to the crank $O\,C$. If the tangential component T be set off along the corresponding crank direction, a *polar* curve of crank effort will be obtained.

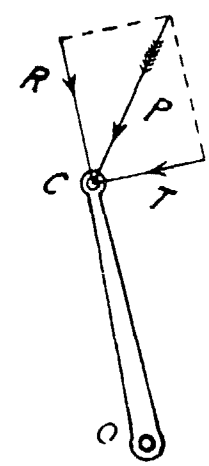


FIG. 230.

If the pressure, P, be constant during the down-stroke, and be directed vertically downwards, the polar curve of crank effort will be a circle. Let p be the effort exerted by the rider at any

instant at his knee-joint in the direction of the motion of the latter, let t be the corresponding tangential effort on the pedal, let s be a very small space moved through by the pedal, and s^1 the corresponding space moved through by the knee-joint. Then the work done at the knee-joint is ps^1, the corresponding work done at the pedal ts; these two must be equal, presuming there is no appreciable loss in the transmission. Therefore

$$t = \frac{s^1}{s} p \qquad \qquad (2)$$

But $\frac{s^1}{s}$ is the ratio of the speeds of the knee-joint and pedal respectively, and is represented by the intercept Df (fig. 21). If, therefore, the effort at the knee-joint be constant during the down-stroke of the pedal, figure 23 is the curve of crank effort as well as the speed curve of the knee.

If, starting from any position, the distance moved through by the pedal relative to the machine be set off along a horizontal line, and the corresponding tangential effort on the crank be erected as an ordinate, a *rectangular* curve of crank effort will be obtained.

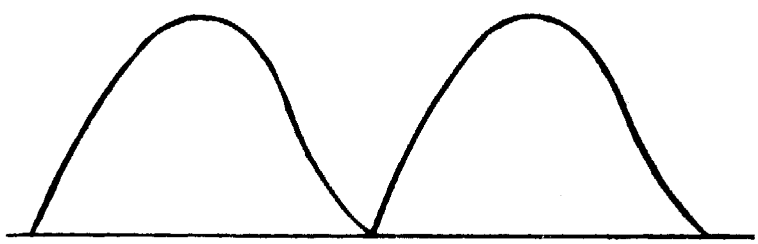


FIG. 231.

Corresponding to the circle as the polar curve of crank effort, the rectangular curve will be a *curve of sines*. Figure 231 shows the rectangular curve corresponding to the down-stroke polar curve in figure 23.

The area included between the base line and the rectangular curve of crank effort represents the amount of work done. The mean height of the rectangular curve therefore represents the mean tangential effort to be applied at the end of the crank in order to overcome the resistance of the cycle.

214. **Actual Pressure on Pedals.**—The actual pressure on the pedal during the motion of the cycle is not even approximately constant. Mr. R. P. Scott investigated the actual pressure on the pedal by means of an instrument which he calls the 'Cyclograph,' the description of which we take from 'Cycling

Art, Energy, and Locomotion.' "A frame, *A A* (fig. 232), is provided with means to attach it to the pedal of any machine. A table, *B*, supported by springs, *E E*, has a vertical movement through the frame *A A*, and carries a marker, *C*. The frame carries a drum, *D*, containing within mechanism which causes it to revolve regularly upon its axis. The cylindrical surface of this drum *D* is wrapped with a slip of registering paper removable at will. When we wish to take the total foot pressure, the cyclograph is placed upon the pedal and the foot upon the table. The drum having been wound and supplied with the registering slip, and the marker *C* with a pencil bearing against the slip, we are ready to throw the trigger and start the drum, by means of a string attached to the trigger, which is held by the rider so that he can start the apparatus at just such time as he desires a record of the pressure."

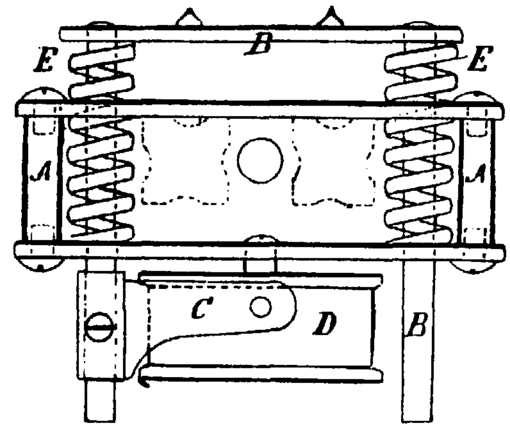


FIG. 232.

Figure 233 shows a cyclograph from a 52-inch 'Ordinary' on a race track, speed 18 miles per hour; figure 234 that from the same

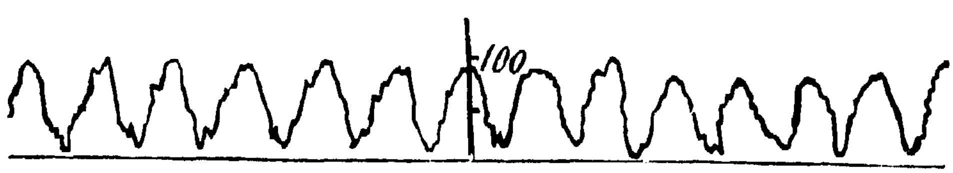


FIG. 233.

machine ascending a gradient 1 in 10, speed 4 miles an hour; and figure 235 is from the same machine back-pedalling down a

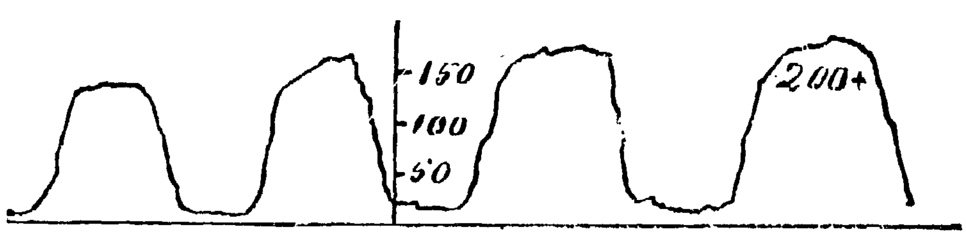


FIG. 234.

gradient 1 in 12. Figure 236 is from a rear-driver geared to 54 inches up a gradient 1 in 20 at a speed of 9 miles an hour; and

figure 237 is from the same machine going up a gradient of 1 in 7 at a speed of 10 miles per hour. The figures on the diagrams

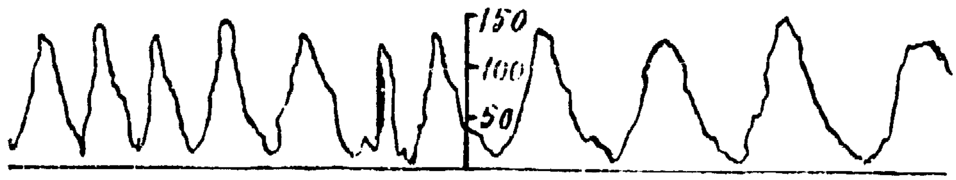


FIG. 235.

are lbs. pressure on the pedal. These curves and many others are discussed in the work above referred to.

These curves give no notion as to the varying *tangential* effort

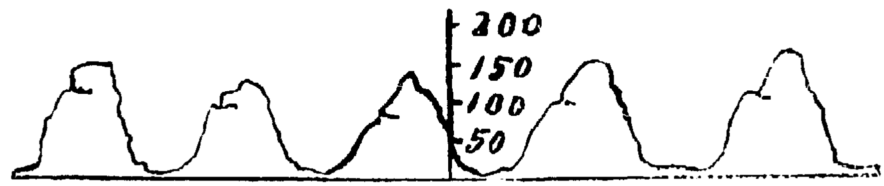


FIG. 236.

on the crank, which is, of course, of more importance than the total pressure. Mallard & Bardon's dynamometric pedal, referred

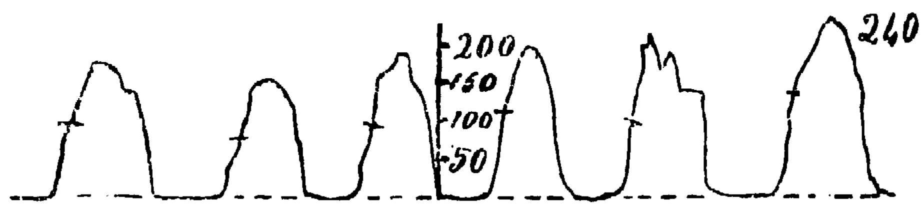


FIG. 237.

to by C. Bourlet, is an instrument in which the tangential component of the pedal pressure is measured and recorded.

215. **Pedalling.**—A vertical push during the down-stroke of the pedal is the most intense effort that the cyclist can communicate, and unfortunately it is the only one that many cyclists are capable of exerting. From Scott's cyclograph diagrams it will be seen that in only one case is the pedal pressure zero during the up-stroke. The first improvement, therefore, that should be made in pedalling is to lift the foot during the up-stroke, though not actually allowing it to get out of contact with the pedal. Toe-clips will be of advantage in acquiring this.

Next, just before the crank reaches its upper dead-centre a horizontal push should be exerted on the pedal, and before it reaches the lower dead-centre the pedal should be clawed backwards. These motions, if performed satisfactorily, will considerably extend the arc of driving.

Ankle Action.—To perform these motions satisfactorily the ankle must be bent inwards when the pedal is near the top, and fully extended when near the bottom. Figures 238, 239, and 240, from a booklet describing the 'Sunbeam' cycles issued by Mr. John Marston, show the positions of the ankle when the crank is at the top, the middle of the down-stroke, and the bottom respectively. The method of acquiring a good ankle action is well described in the 'Sunbeam' booklet and in Macredy's 'The Art and Pastime of Cycling.' Besides increasing

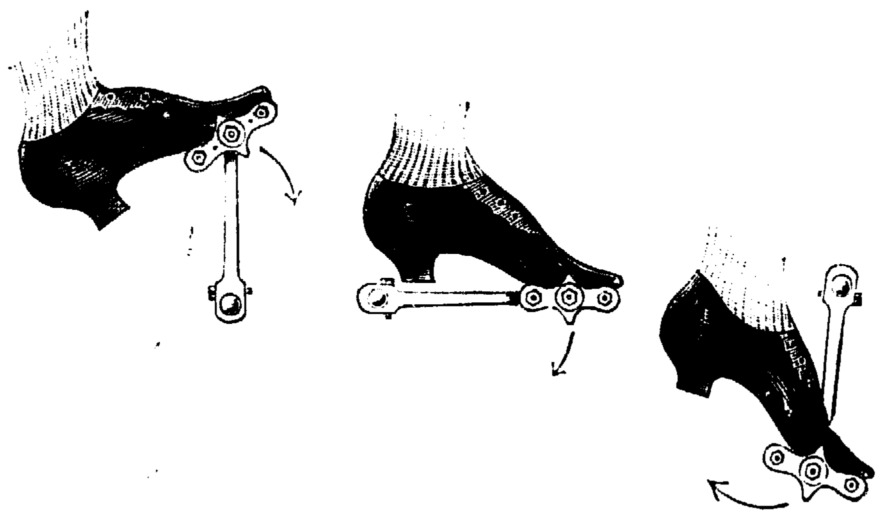


FIG. 238. FIG. 239. FIG. 240.

the arc of driving, ankle action has the further advantage of diminishing the extent of the motion of the leg. With a good ankle action the speed curves shown in figures 23, 501, and 511 may be considerably modified; in fact, the addition of a fifth link (between the foot and ankle) to the kinematic chain in figure 22 makes the motion of the leg indeterminate.

If the shoe of the rider be fastened to the pedal an upward pull may be exerted, and the action of pedalling becomes more like that of turning a crank by hand, the arc of action being extended to the complete revolution. With *pulling pedals* more work is thrown on the flexor muscles of the legs, to the corresponding relief of the extensors.

216. **Manumotive Cycles.**—A few cycles, principally tricycles, have been designed to be driven by the action of the hand and arms.

Singers' 'Velociman' has been for a number of years the best example of this type of machine. Figure 241 shows an up-to-date example. The effort is applied by the hands to two long

levers, which, by sliding joints in place of connecting-rods, drive cranks at opposite ends of an axle ; this axle is connected by chain

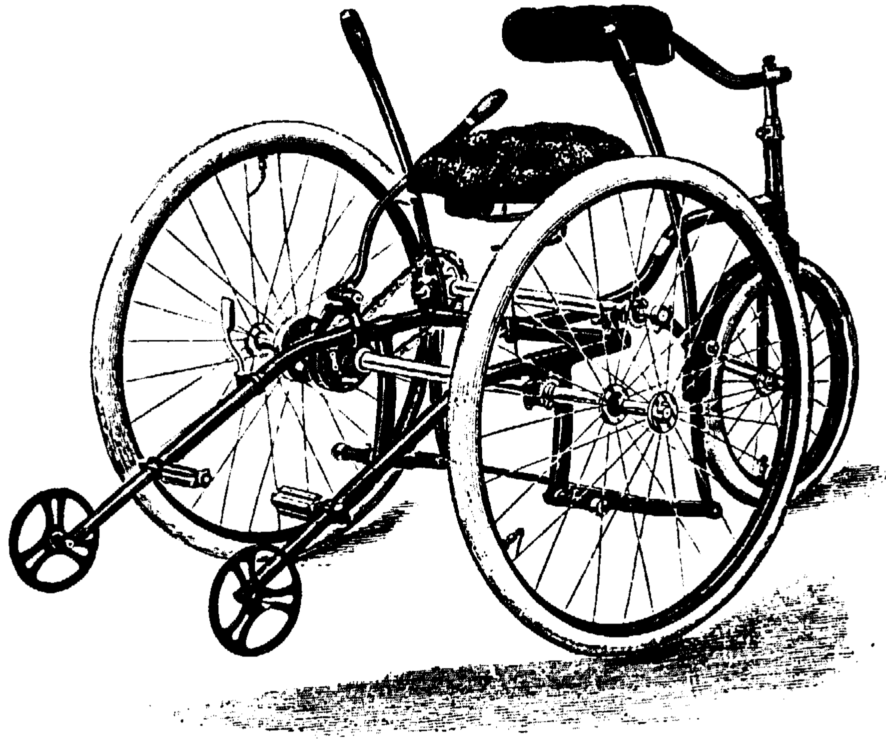


FIG. 241.

gearing to the balance gear on the driving-axle. The steering is done by the back pressing against a cushion supported at the end of a long steering bar.

217. **Auxiliary Hand-power Mechanisms.**—A number of cycles have been made from time to time with gearing operated by hand, having the intention of supplementing the effort communicated by the pedals. The idea of the inventors is that the greater the number of muscles concerned in the propulsion, the greater will be the speed, or a given speed will be obtained with less fatigue ; but though this may be true for extraordinary efforts of short duration, it is probably quite erroneous for long-continued efforts. Whatever set of muscles be employed to do work, a man has only one heart and one pair of lungs to perform the functions required of them. It is a matter of everyday experience that the cyclist can tax his heart and lungs to their utmost, using only pedals and cranks ; so that, unless inventors can provide a method of stimulating these organs to do more than they are at present capable of, it seems worse than useless to complicate the machine with auxiliary hand-power mechanism. Re-

garded as a motor, the human body may be compared to a number of engines deriving steam from one boiler, supplied with feed-water by one feed-pump. If one engine is capable of using all the steam generated in the boiler, no additional, but rather less, useful work will be obtained by setting additional engines running. It is a fact well known to engineers that a steam-engine works most economically when running under its heaviest load. One engine, therefore, will utilise the steam generated in the boiler more efficiently than several. The lungs may be compared to the furnace of the boiler, the blood to the feed-water, the heart to the feed-pump which circulates the feed-water, the muscles of the legs to an engine capable of utilising all the energy supplied by the combustion of the fuel in the furnace, the arms to a small engine. If the analogy can be pushed so far, less work will be got from the body by using both legs and arms simultaneously than by using the legs only ; and this quite independently of the frictional resistance of the additional mechanism.

The 'Road-sculler' and 'Oarsman' tricycles were designed so that the rider might exercise the muscles of his legs, back, chest, and arms, as in rowing. The speed attained was less than in the crank-driven tricycle, the mechanism being more complex and therefore less efficient, while from the foregoing discussion it seems probable that the rider, though using more muscles, actually developed less indicated power.

PART III

DETAILS

PART III

DETAILS

CHAPTER XXII

THE FRAME (DESCRIPTIVE)

218. **Frames in General.**—The frame of a bicycle forms practically a beam which carries a load—the weight of the rider—and is supported at two points, the wheel centres. In order to allow of steering, this beam is divided into two parts connected by a hinge joint—the steering-head. The two parts are sometimes referred to as the 'front-frame' and the 'rear-frame'; the front-frame of a 'Safety' including the front fork, head-tube, and the handle-bar. The rear-frame has assumed many forms, which will be discussed in some detail. In all bicycles that have attained to any degree of success the rear-frame has been the larger of the two; hence sometimes when 'the frame' is mentioned without any further qualification, the rear-frame is meant. It is usually evident from the context whether 'the frame' means the rear-frame or the complete frame.

219. **Frames of Front-drivers.**—The 'Ordinary' has the simplest, structurally, of all cycle frames, consisting of a single tube, called the backbone, forked at its lower extremity for the reception of the hind wheel, and hinged to the top of the fork carrying the front wheel. The frame of the 'Geared Ordinary' is the same as that of the 'Ordinary,' the distance between the seat and the top of the driving-wheel being too small to admit of bracing the structure. With the further reduction of the size of the driving-wheel, and the greater distance obtained between the saddle and top of the driving-wheel, it becomes possible to use a braced frame. Figure 242

shows a front-driving frame made by the Abingdon Works Company (Limited). Here the weight of the rider is taken up by the two straight tubes, each of which will be subject to bending-moment due to half the total weight.

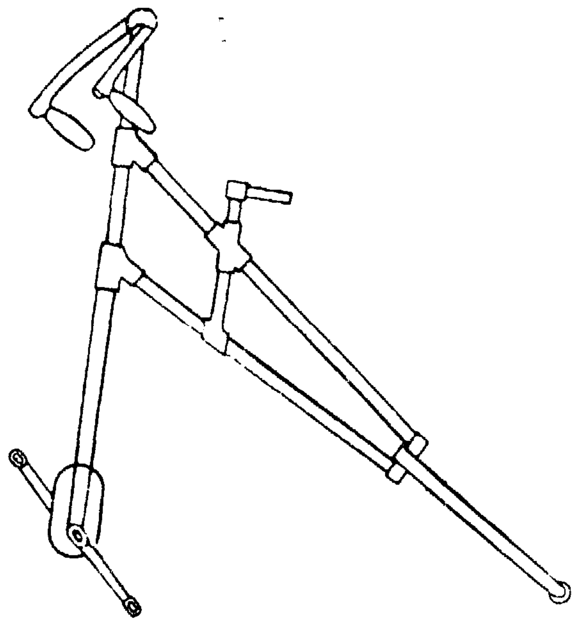


FIG. 242.

Figure 132 shows one form of frame used by the Crypto Works Company (Limited), in their 'Bantam.' The bracing in this is more apparent than real, since the weight of the rider is transferred to the middle of a straight tube of very little less length than the total distance between the wheel centres. This tube must, therefore, be made strong enough to resist the bending-moment.

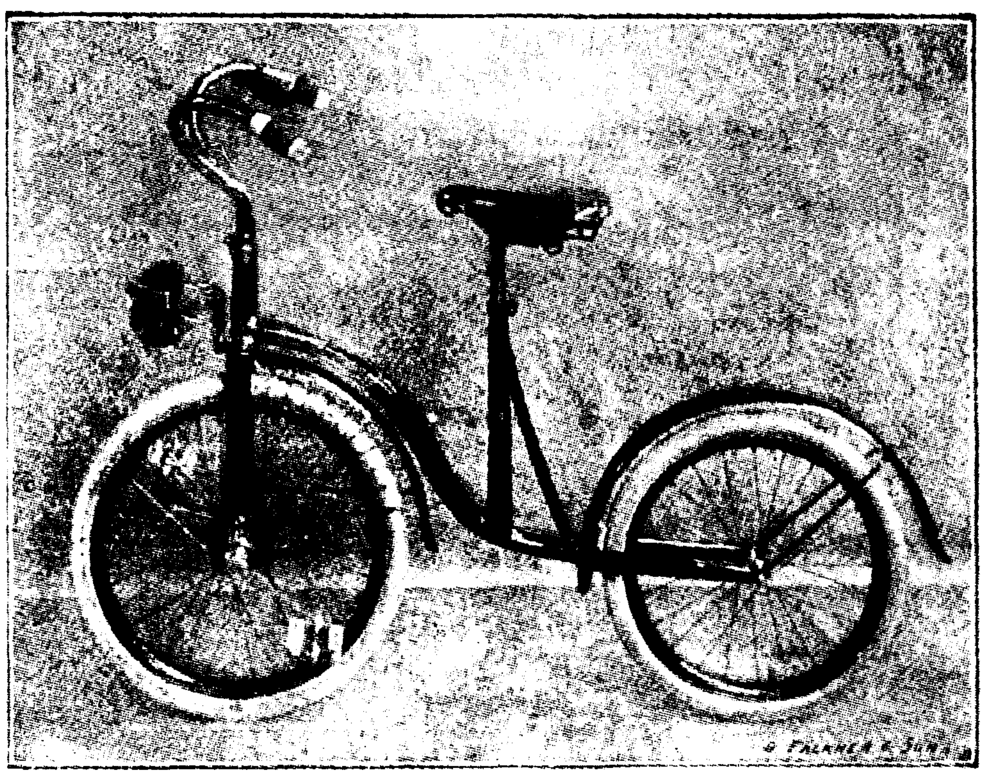


FIG. 243.

Figure 243 shows the frame of the 'Bantamette,' made by the same company, and which can be ridden by a lady with skirts.

Here, of course, the backbone is subjected to bending-stresses, and a very strong tube must be used for it. Figure 291 shows a properly braced front-driving frame designed by the author, which is practically equivalent to a triangular truss. The short tube joining the steering-head to the seat-lug is made stout enough to resist the bending due to the saddle-pin attachment, while the seat-struts are subjected only to compression, and the lower stays to tension.

220. **Frames of Rear-drivers.**—The rear-driving chain-driven 'Safety' introduced in 1885 is kinematically the same as the popular machine of the present day. The greatest difference between them lies in the design of the frame. So many designs of frame have been used that we can only notice a few general types here.

The original 'Humber' frame (fig. 128) has a general resemblance to the present-day diamond-frame, though from a structural point of view, the want of a tube joining the saddle-pillar to the crank-axle makes it greatly different as regards strength.

Figure 244 shows the 'Pioneer' dwarf Safety, made by H. J. Pausey, 1885. This is of the cross-frame type, and consists practically of two members, one joining the driving-wheel spindle to the steering-head, the other running from the saddle to the crank-axle. It will be noticed that the frame is not braced or stayed in any manner, so that the whole weight of the rider is transferred to the backbone. When driving, the pull of the chain tends to bring the crank-axle and driving-wheel centres nearer together, and there being no direct struts to resist this action, the frame is structurally weak. In this respect it is much worse than the 'Humber' frame (fig. 128).

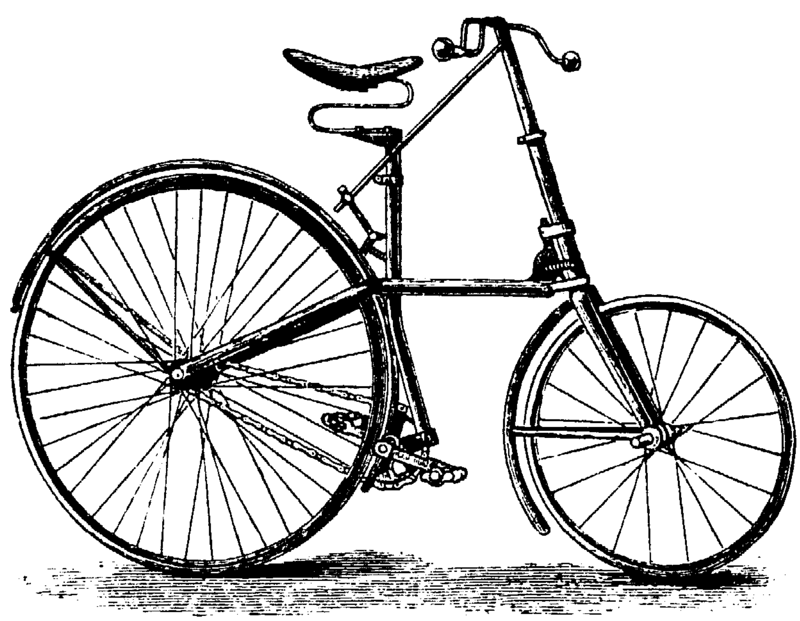


FIG. 244.

Figure 126 shows the 'Rover' Safety made by Messrs. Starley & Sutton, 1885. The frame is of the open diamond type, the

front fork is vertical, and the steering is not direct, but the handle-bar is mounted on a secondary spindle connected by short links to the front fork.

Figure 245 shows a Safety made by the Birmingham Small Arms and Metal Co., Limited. The principal difference between

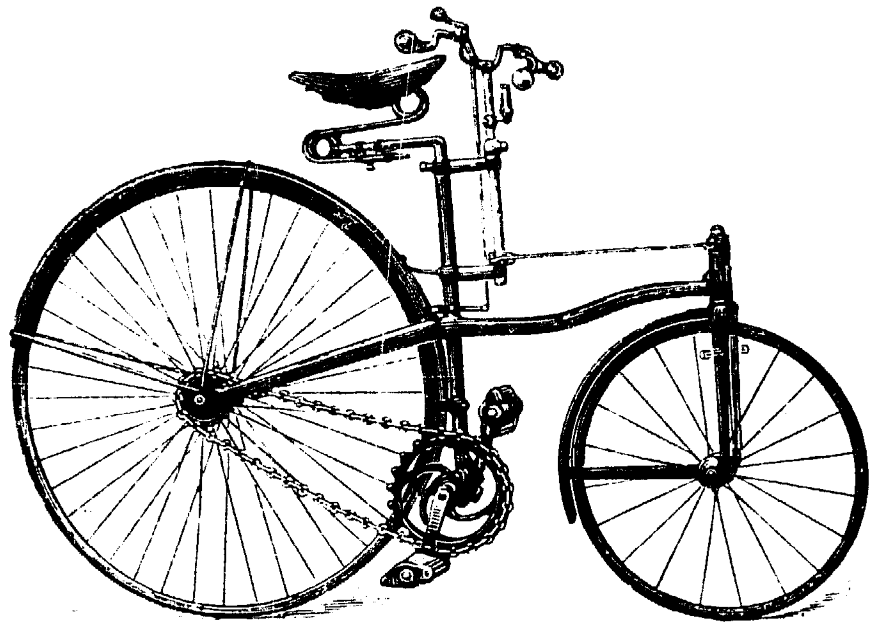


FIG. 245.

this frame and that of figure 244 consists in the substitution of indirect for direct steering.

Figure 127 shows the 'Rover' Safety, made by Messrs. Starley

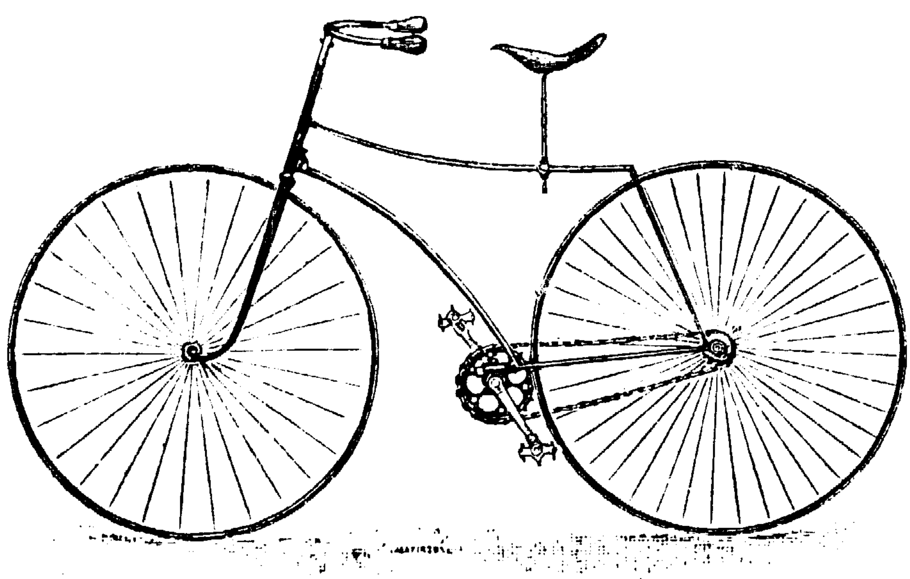


FIG. 246.

& Sutton in 1886. The frame is of the open diamond type, with curved tubes, and direct steering is used. The approximation

to the present type of frame is closer than in any of the previous examples.

Figure 246 shows the 'Swift' Safety, made by the Coventry

FIG. 247.

Machinists Co., 1887. This frame is of the open diamond type; the top and bottom tubes from the steering head are curved.

The first improvement on the elementary cross-frame (fig. 244) was to insert struts, or a lower fork, between the crank-bracket

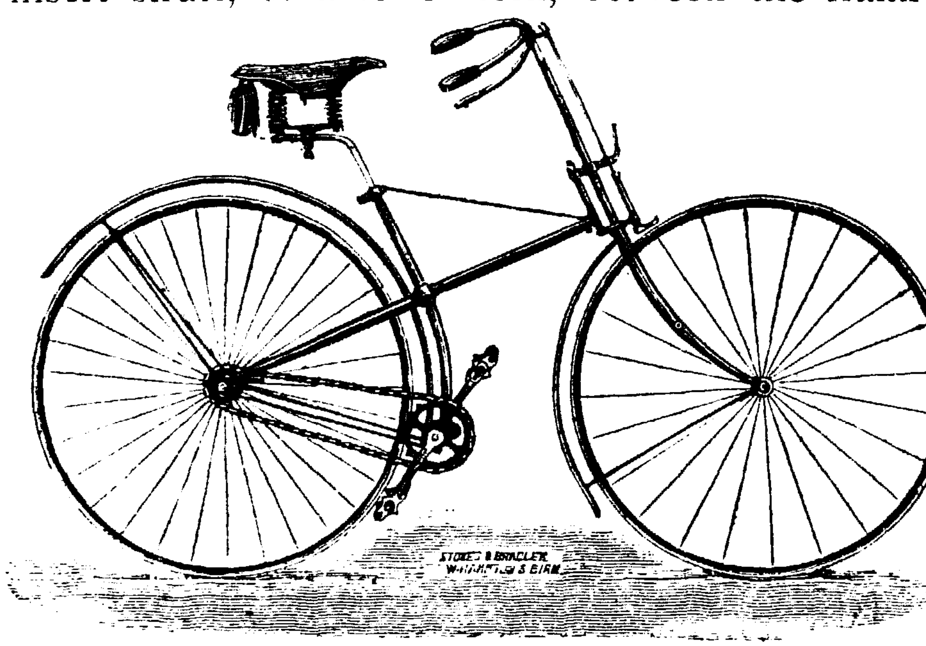


FIG. 248.

and driving-wheel spindle (fig. 247), so that the pull of the chain could be properly resisted. Another improvement was to connect the steering-head and the top of the saddle-post by a light stay (fig. 248). In the 'Ivel' Safety of 1887 (fig. 249) a stay ran from the steering-head to the crank-bracket, but the chain-struts were

omitted. The 'Humber' Safety of this period (fig. 250) had the crank-bracket stay and chain-struts. The 'Invincible' Safety, made by the Surrey Machinists Co. in 1888 (fig. 251), had, in

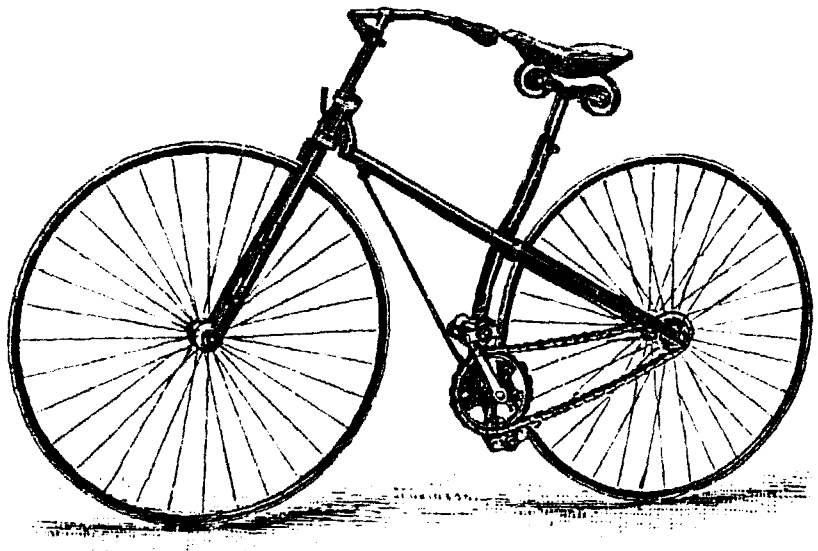


FIG. 249.

addition, a stay between the steering-head and top of saddle-post; while a later machine (fig. 252), by the same firm, had stay-rods from the driving-wheel spindle to the top of saddle-post. This

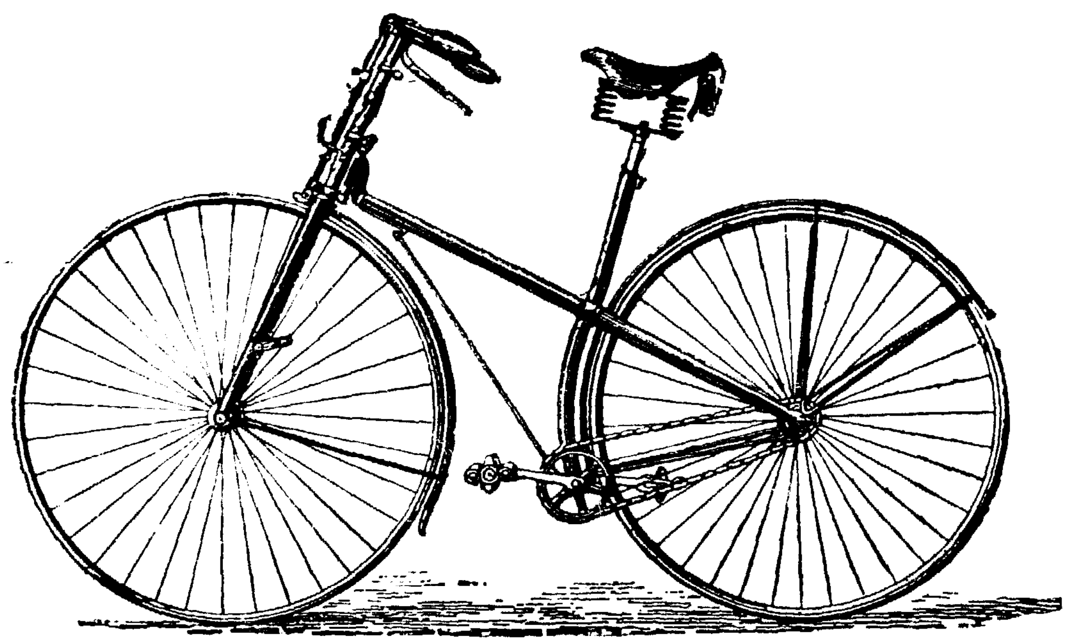


FIG. 250.

bicycle was made forkless, the wheel-spindles being supported only at one end; but in this respect the design is not to be recommended.

The frame of the 'Sparkbrook' Safety, 1887 (fig. 253), may be

noticed. It is a kind of compromise between the cross-frame and the open diamond ; the crank-bracket and driving-wheel spindle

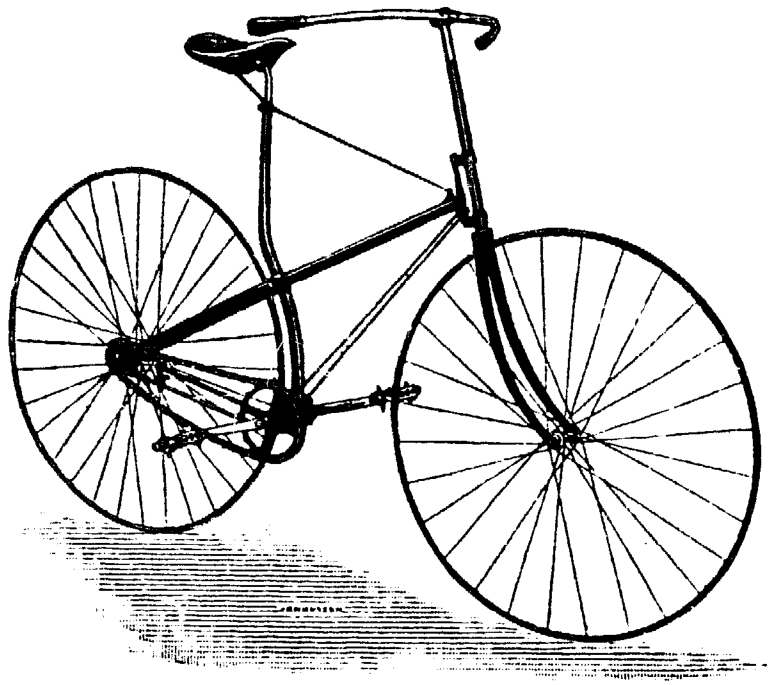


FIG. 251.

are directly connected, but the crank-axle is connected to a point about the middle of the upper tube of the frame. The bending-

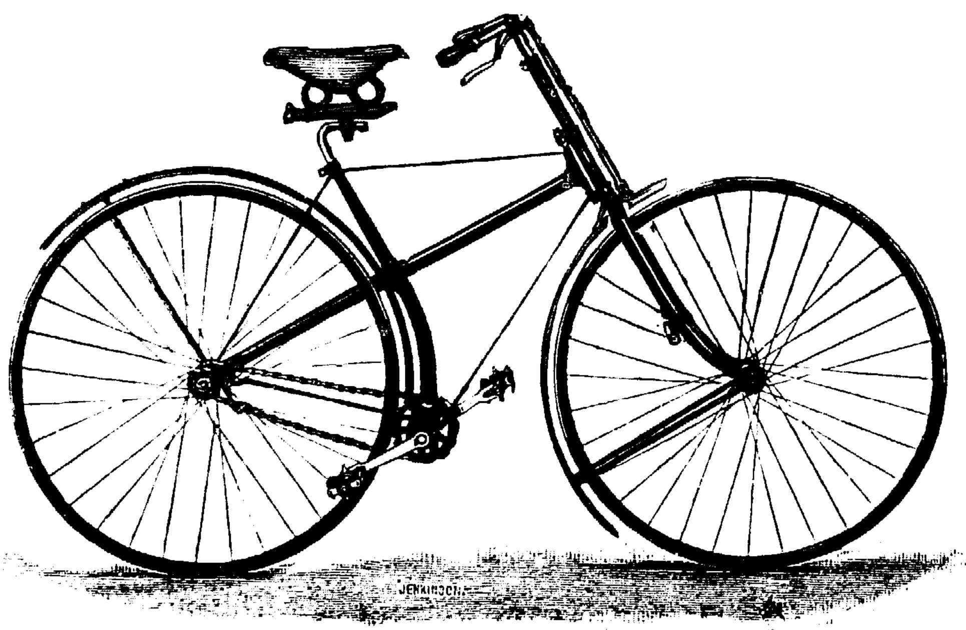


FIG. 252.

moment, which attains nearly its maximum value at this point, is resisted by this single tube, which consequently must be rather heavy.

The frame of the 'Quadrant' bicycle (fig. 254) differs essentially from either the diamond- or cross-frame. In this bicycle the main frame is continued forward on each side of the steering-

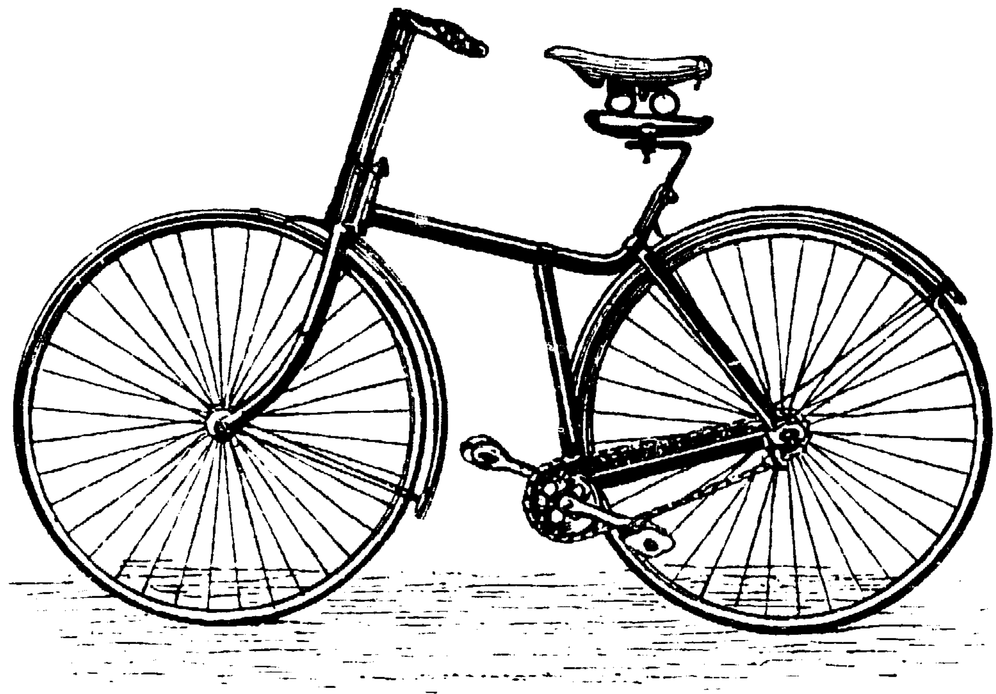


FIG. 253.

wheel; the spindle of the steering-wheel is not held in a fork, but its ends are mounted on cases which roll on curved guides or 'quadrants.' From each case a light coupling-rod gives connec-

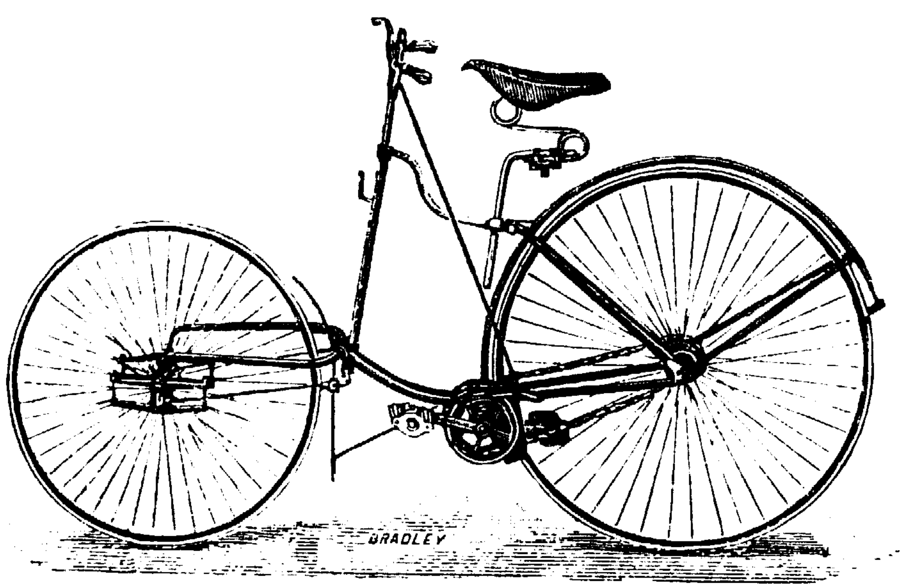


FIG. 254.

tion to a double lever at the bottom of the steering-pillar. The frame in front of this steering-pillar consists practically of two tubes with no bracing, while the bracing of the rear portion is

very imperfect. This arrangement for controlling the motion of the steering-wheel is the same as used in the 'Quadrant' tricycle.

The frame of the 'Rover' Safety of 1888 (fig. 255) shows a

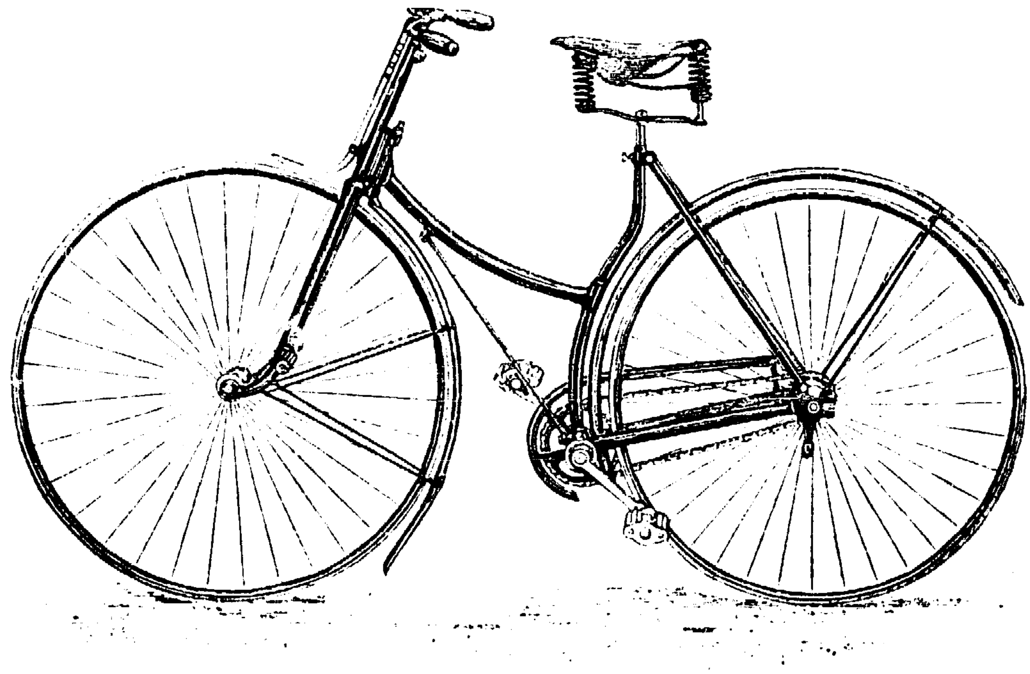


FIG. 255.

great advance on any of the earlier frames above described. It may be described as a combination of the cross- and diamond-

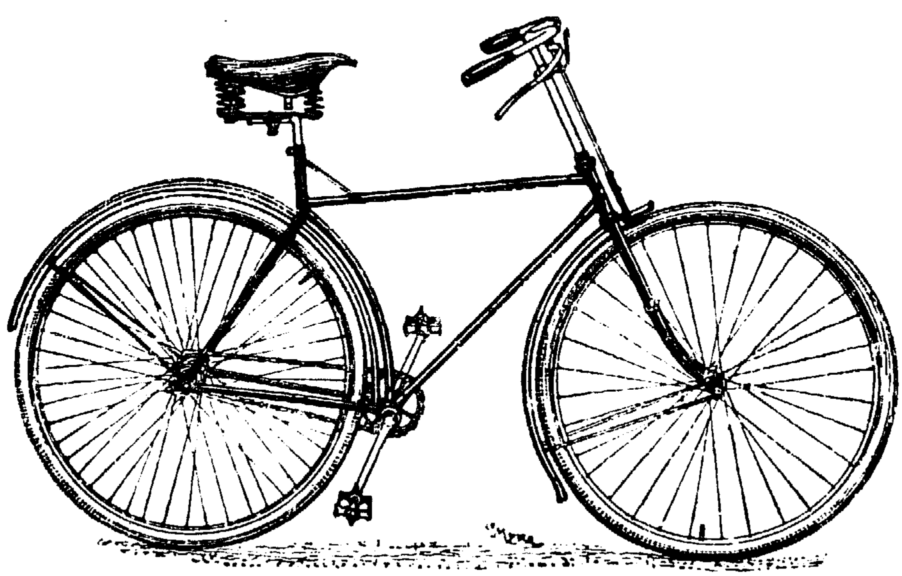


FIG. 256.

frames. The main tube from the steering-head is joined on about the middle of the down-tube from the saddle to the crank-bracket, which thus may be considered to be supported at its ends and loaded in the middle, and must therefore be fairly heavy to resist

the bending-moment on it. Another weak point in the design is the making of the top tube curved instead of straight.

The 'Referee' frame (fig. 256) was one of the earliest with

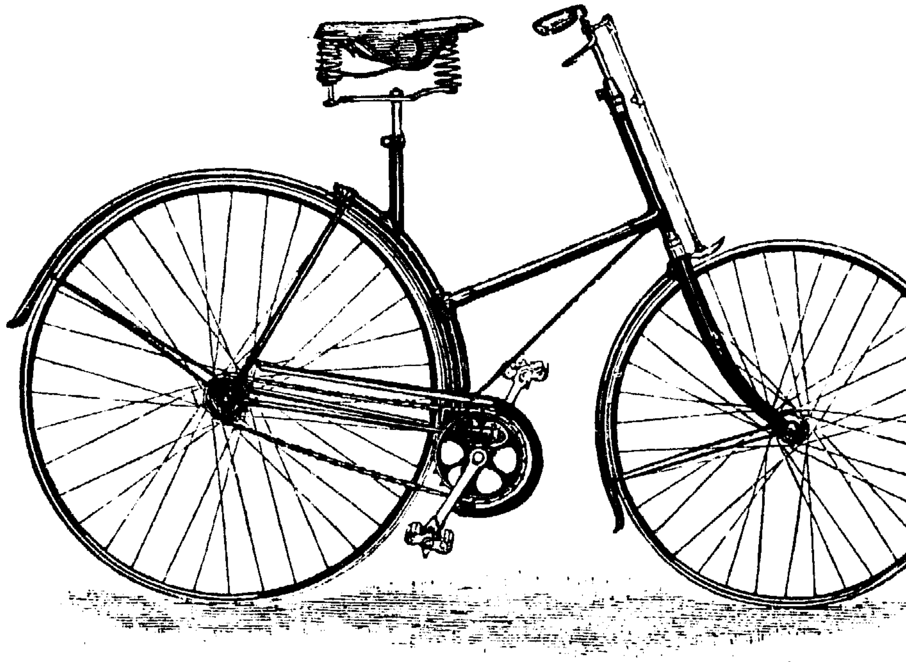


FIG. 257.

practically perfect bracing. The crank-bracket being kept as near as possible to the rim of the driving-wheel, the diamond was stiffened by a curved down-tube. A short vertical saddle-tube was

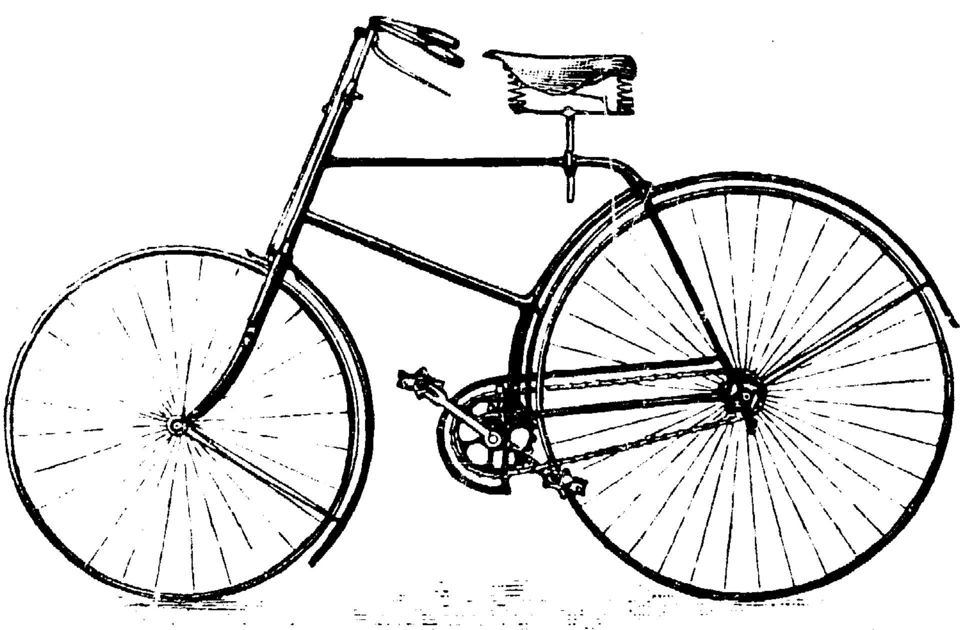


FIG. 258.

continued above the top tube, thus allowing the saddle and pin to be turned forward or backward—a good point which has been abandoned in later frames. Ball-socket steering was used.

Figure 257 shows the Safety made by Singer & Co., 1888, the frame of which differs very little essentially from that of figure 255.

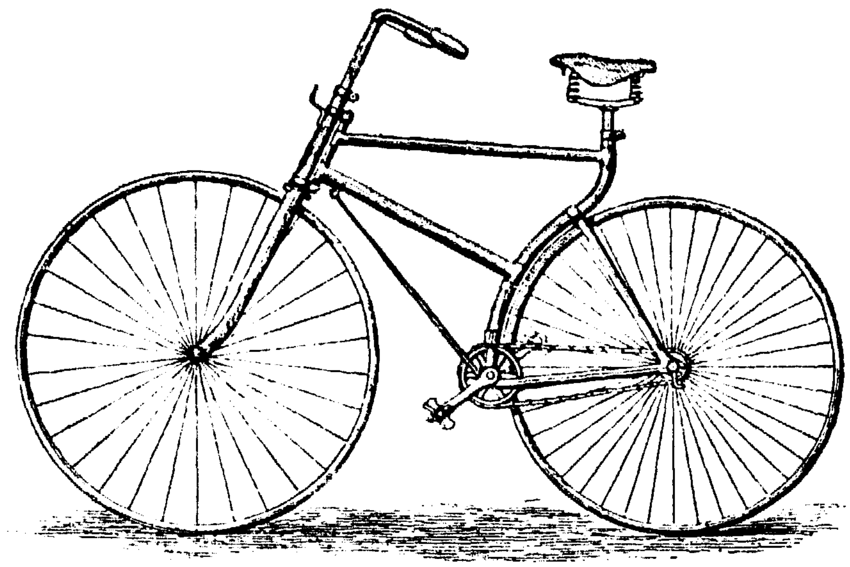


FIG. 259.

Figure 258 shows the 'Singer' Safety of 1889, the frame of which differs considerably from all types hitherto described. The

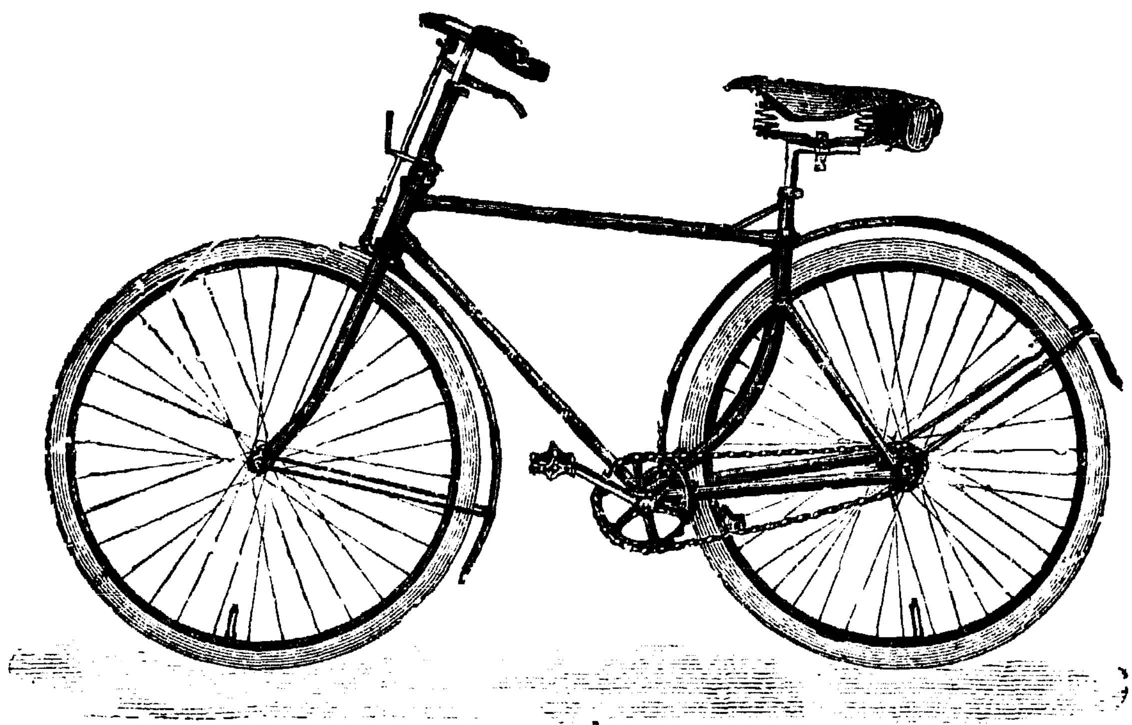


FIG. 260.

remarks applied to the design of the frame in figure 255 may also be applied to this frame.

The 'Ormonde' (fig. 259) and the 'Mohawk' (fig. 260) frames may be noticed, the latter having the down-tube from saddle to crank-bracket in duplicate.

Figure 130 shows the 'Humber' Safety of 1889. This frame gives the first close approximation to the present almost universally used 'Humber' frame.

In 1890 the 'Humber' Safety, with extended wheel-base, was introduced. In this machine the distance between the crank-axle and the driving-wheel was increased, thereby increasing the distance between the points of contact of the two wheels with the ground. With this increased distance it was possible to join the seat-lug to the crank-bracket by a straight down-tube, thereby giving the well-known 'Humber' frame (fig. 131). The stem of the saddle-pin goes inside this tube, and a neater appearance is obtained thereby. This frame is not a perfectly braced structure, the introduction of a tube to form one of the diagonals of the diamond being necessary to convert it into a perfectly framed structure. This has been done in the 'Girder' Safety frame (fig. 296). With a well designed 'Humber' frame, however, the possible bending-moment on the tubes, due to the omission of the diagonal, is so small that it is practically not worth while to introduce the extra tube.

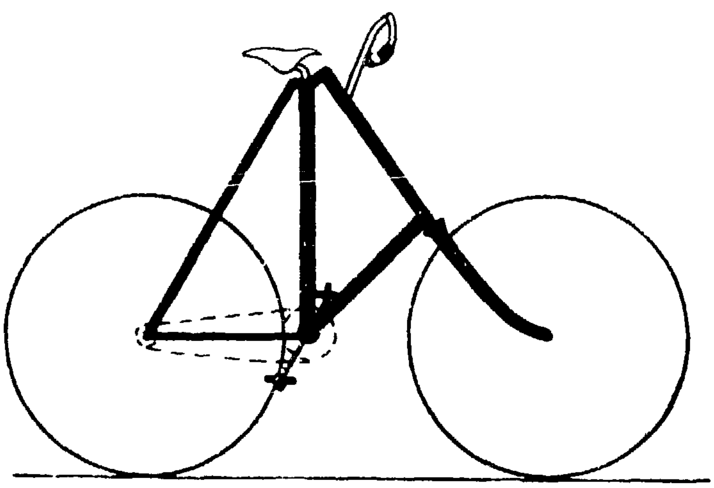


FIG. 261

Quite recently a 'pyramid'-frame (fig. 261) has been introduced in America. It remains to be seen whether the excessive rake of the steering head, necessary with this design, will allow of the easy steering we are accustomed to with the diamond-frame.

Bamboo Frames.—From the discussion of the stresses on the frame (chap. xxiii.) it will be seen that when the frame is properly braced, and its members so arranged that the stresses on them are along their axes, the maximum tensile or compressive stress on the material is small. If a steel tube were made as light as possible, with merely sufficient sectional area to resist these principal stresses, it would be so thin that it would be

unable to resist rough handling, and would speedily become indented locally. A lighter material with greater thickness, though of less strength, would resist these local forces better. The bamboo frame (fig. 262) is an effort in this direction, the

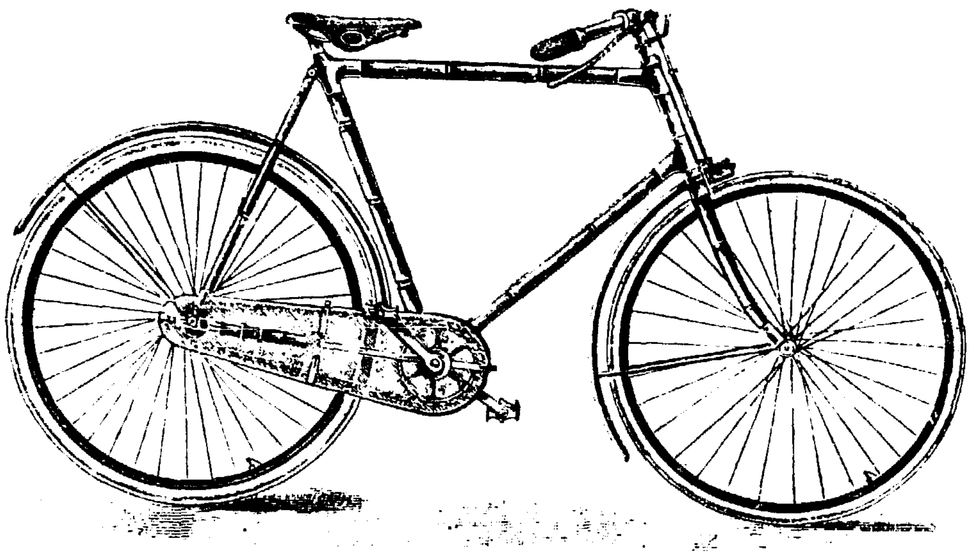


FIG. 262.

bamboo tubes being stronger locally than steel tubes of equal weight and external diameter.

Aluminium Frames.—From the extreme lightness of aluminium compared with iron or steel, many attempts have been made to employ it in cycle construction. The tenacity of the pure metal is, however, very low, and its ductility still lower, compared with steel; while no alloy containing a large percentage of aluminium, and therefore very light, has been found to combine the strength and ductility necessary for it to compete favourably with steel. Of course, for parts which are not subjected to severe stresses it may probably be used with advantage.

221. **Frames of Ladies' Safeties.**—The design of the frame of a Ladies' Safety is more difficult than the design of the frame for a man's Safety. In the early Ladies' Safeties the frame was usually of the single tube type, and may be represented by the 'Rover' Ladies' Safety (fig. 263). The single tube from the crank-bracket to the steering-head is subjected to the entire bending-moment, and must therefore be of fairly large section. If the lady rider wears skirts, the top tube, as used in a man's bicycle, must be omitted; and if a second bracing tube be introduced, it must be very low down. Figure 264 shows one of the usual

forms of Ladies' Safety, a tube being taken from the top of the steering-head to a point on the down-tube a few inches above the crank-bracket. By this arrangement, of course, the down-

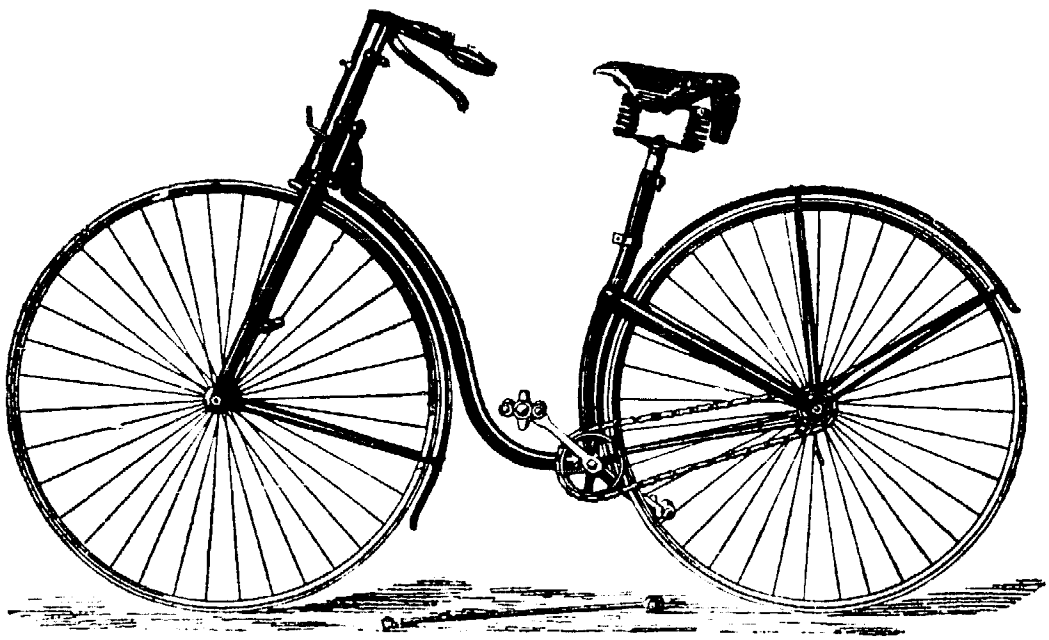


FIG. 263.

tube is subjected to a bending stress, while the frame, as a whole, is weakest in the neighbourhood of the crank-axle. Since the bending-moment on the frame diminishes from the crank-axle

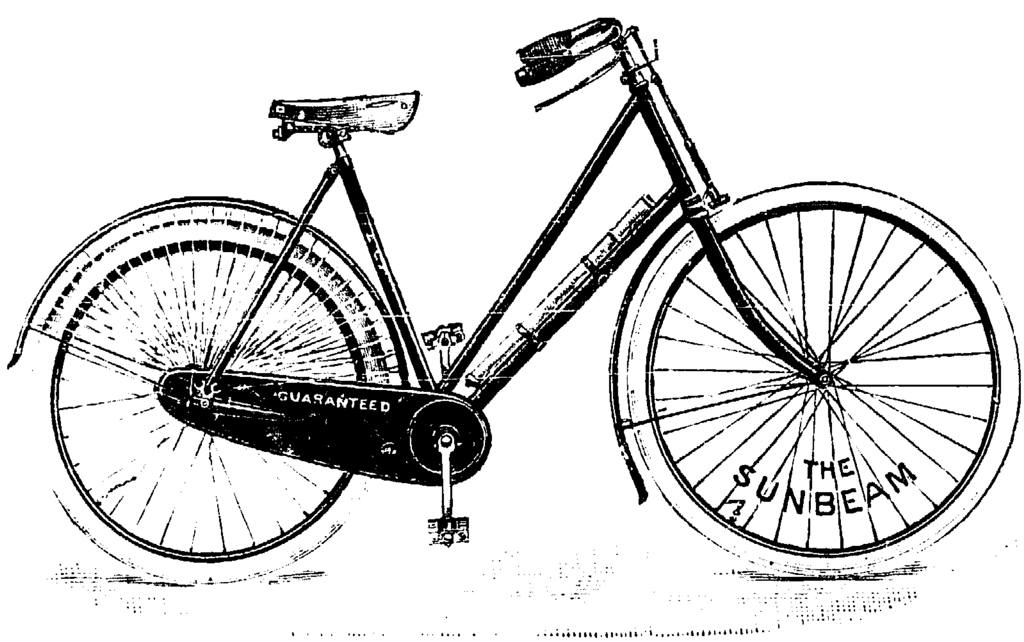


FIG. 264.

towards the front wheel centre, it is better to have the two tubes from the steering-head diverging (fig. 265) instead of converging as they approach the crank-axle ; the depth of the frame would

then vary proportionally to the depth of the bending-moment diagram, and the bending stresses on the members of the frame

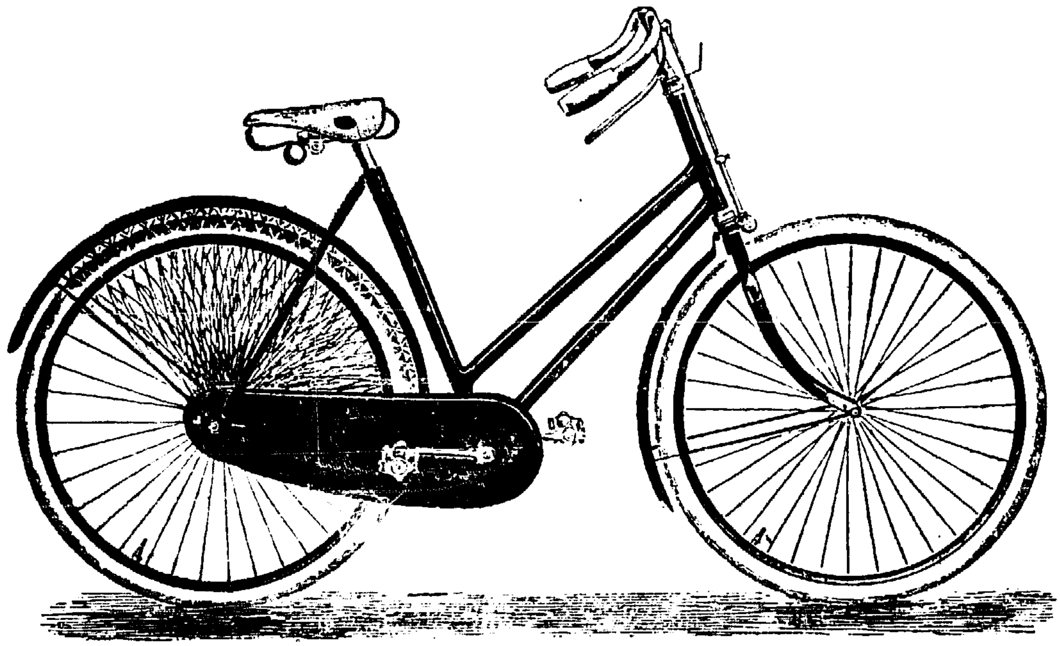


FIG. 265.

would be least. Such an attempt at bracing the frame of a Ladies' Safety, as is illustrated in figure 266, is useless, since at the point P the depth of the frame is zero, and the only improvement is that the bending at the point P is resisted by two tubes instead of one.

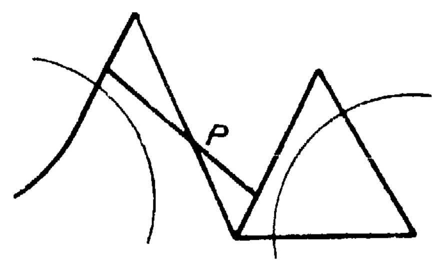


FIG. 266.

222. **Tandem Frames.** — A great variety of frames are in use at present, the processes of natural selection not having gone on for such a long time as is the case with frames for single machines. A frame (fig. 267), resembling that of the ordinary diamond-frame, with the addition of a central parallelogram, has been used. It will be noticed at once that the middle portion is not arranged to the best advantage for resisting shearing-force, so that as regards strength, the middle portion of the frame is simply equivalent to two tubes lying side by side and subjected to bending.

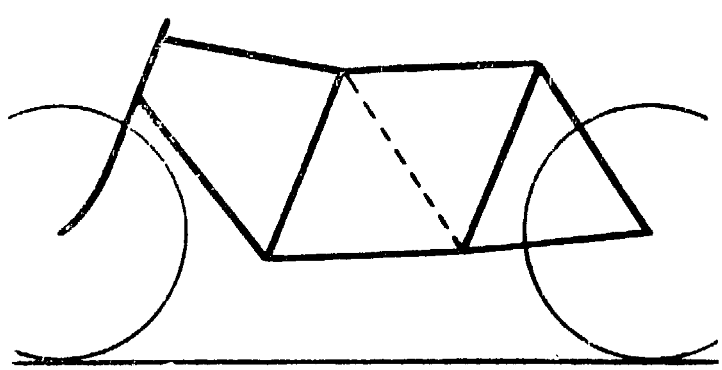


FIG. 267.

Figure 268 shows a tandem frame, by the New Howe Machine Company, in which three tubes resist the bending on any vertical section; and figure 269 shows a frame, by the Coventry Machin-

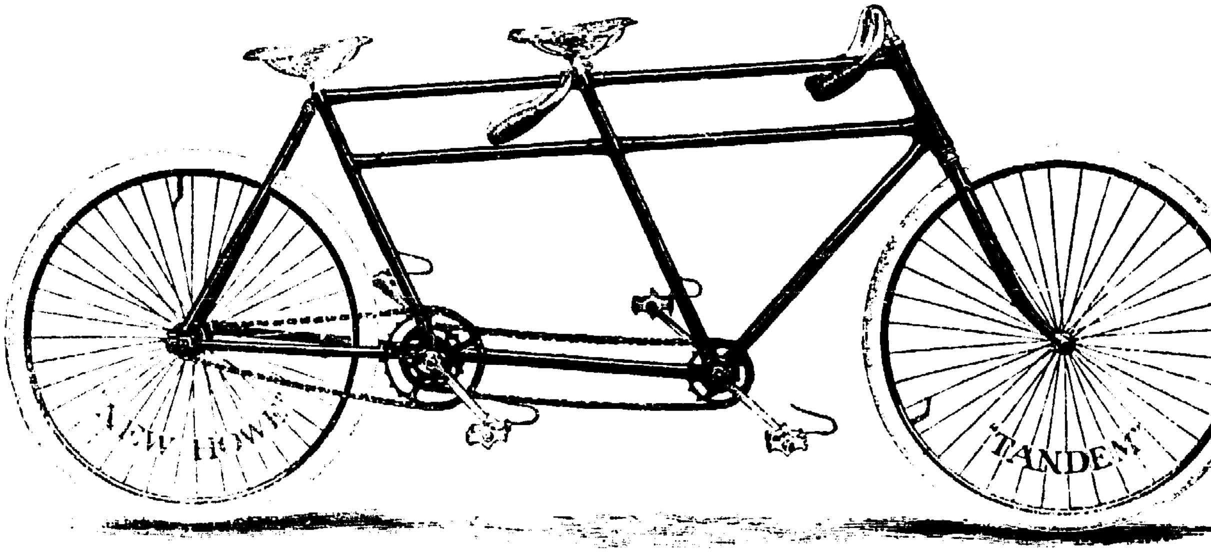


FIG. 268.

ists' Company, with the front seat arranged for a lady. Both these frames should be stronger, weight for weight, than that in figure 267, but they are not perfectly braced structures, and the bending-moment on the tubes will be considerable.

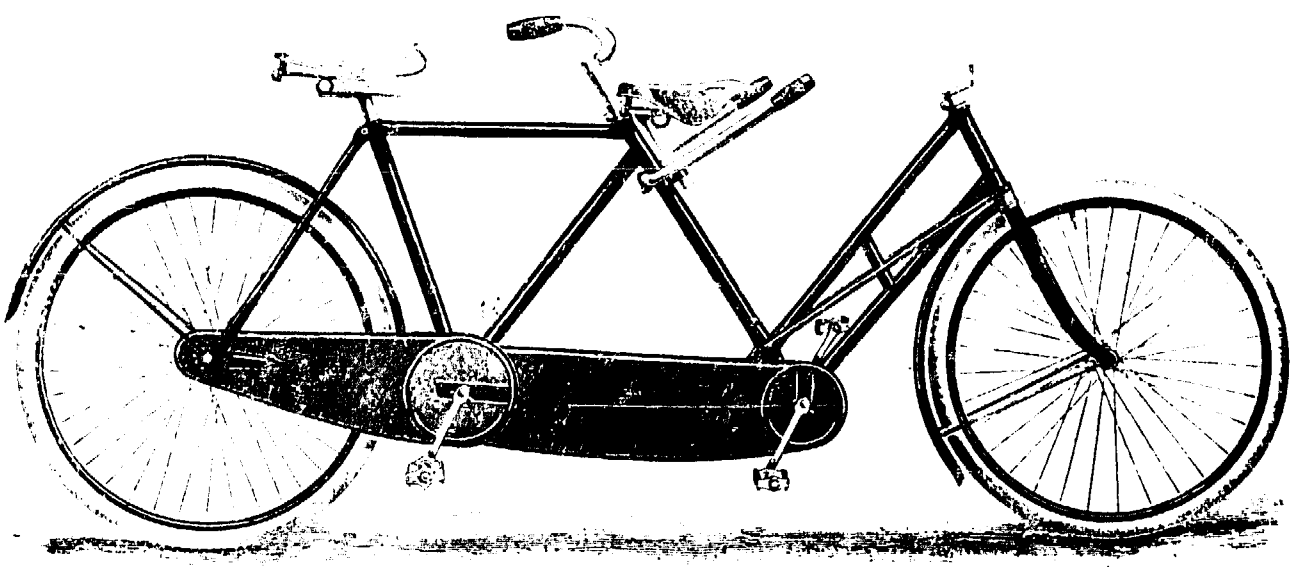


FIG. 269.

The addition of a diagonal to the central parallelogram, indicated by the dotted line (fig. 267), converts the frame into a braced structure, and the strength is proportionately increased.

The front quadrilateral of the frame (fig. 267) requires a diagonal to make the frame a perfectly braced structure, and, though riding along a level road, it is possible, by properly disposing the top and bottom tubes, to insure that there shall be no bending on them, it would seem advisable to provide against contingencies by inserting this diagonal in tandem frames. Such is done in the 'Thompson & James's' frame (fig. 140).

Figure 270 shows a tandem frame, made by Messrs. J. H. Brooks & Co., intended to take a lady on either the front or back seat. On the side of the machine on which the chain is placed, instead of a single fork-tube two tubes are used, one above and one below the chain, and both lying in the plane of the chain.

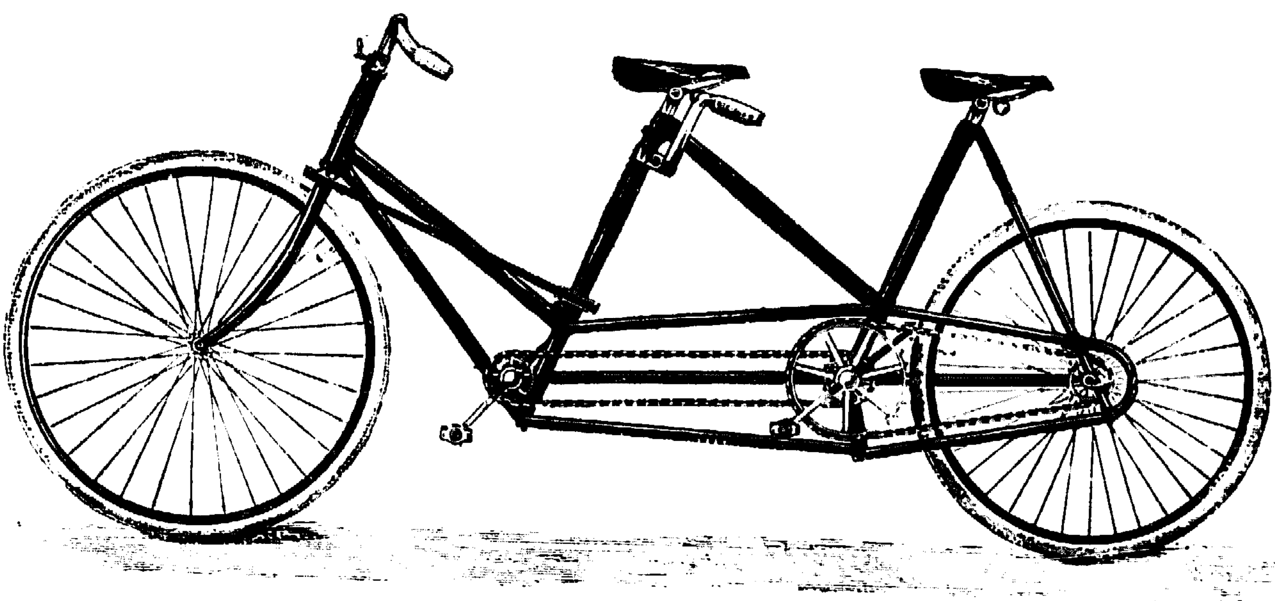


FIG. 270.

Thus the lower part of the frame constitutes a beam to resist the bending-moment, and the upper portions are used merely to support the saddles.

Figure 271 shows a tandem frame also intended to take a lady on either the front or back seat, designed by the author. The frame is dropped below the axle the lower part is, in fact, a braced structure of exactly the same nature as that in figure 267. The crank-bracket is held by a pair of levers, the lower ends of which are hinged on the pin at the lower point of junction of the frame tubes. The upper ends can be clamped in position on the tubes which form the chain-struts. The driving-wheel spindle is

thus permanently fastened to the frame, and therefore remains always in track. A single screw is used to adjust the crank-bracket, on releasing the top clamping screws of the supporting levers. Although this is a reversion to the hanging crank-bracket, it may

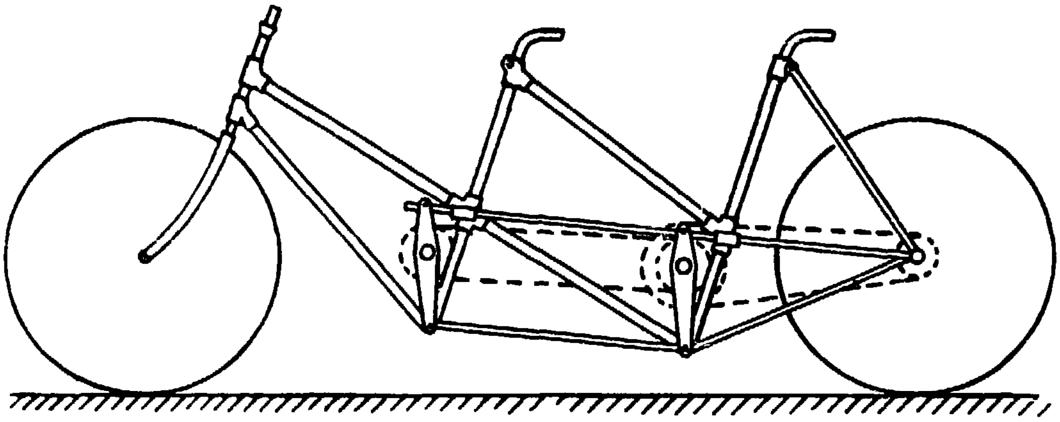


FIG. 271

be pointed out that it is connected rigidly to the frame at four points, and may therefore be depended upon not to work loose.

223. **Tricycle Frames.**—In the early tricycles Y-shaped

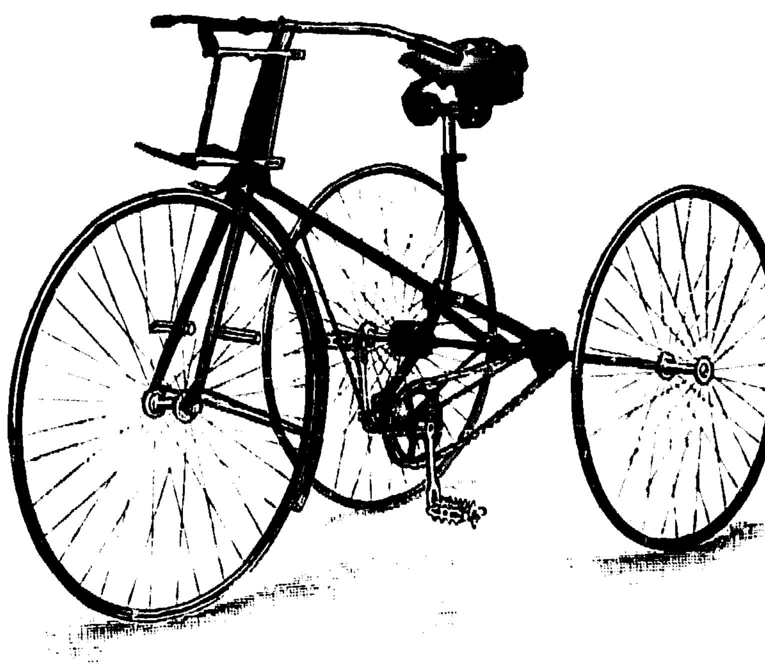


FIG. 272.

frames for front-driving rear-steerers and loop-frames for front-steerers were usually employed, while in side-drivers, such as the Coventry Rotary, the frame was T-shaped, the top of the T being in a longitudinal direction. The frame of the 'Cripper' tricycle (fig. 150) was also T-shaped, the top of the T forming a bridge

supporting the axle, and the vertical branch of the T running forward from the middle of the axle to the steering-head and supporting the crank-axle and seat. These frames were almost entirely unbraced, and their strength depended only on the diameter and thickness of the tubes.

The diamond-frame for tricycles, on the same general lines as the diamond-frame used in bicycles, marks a great improvement

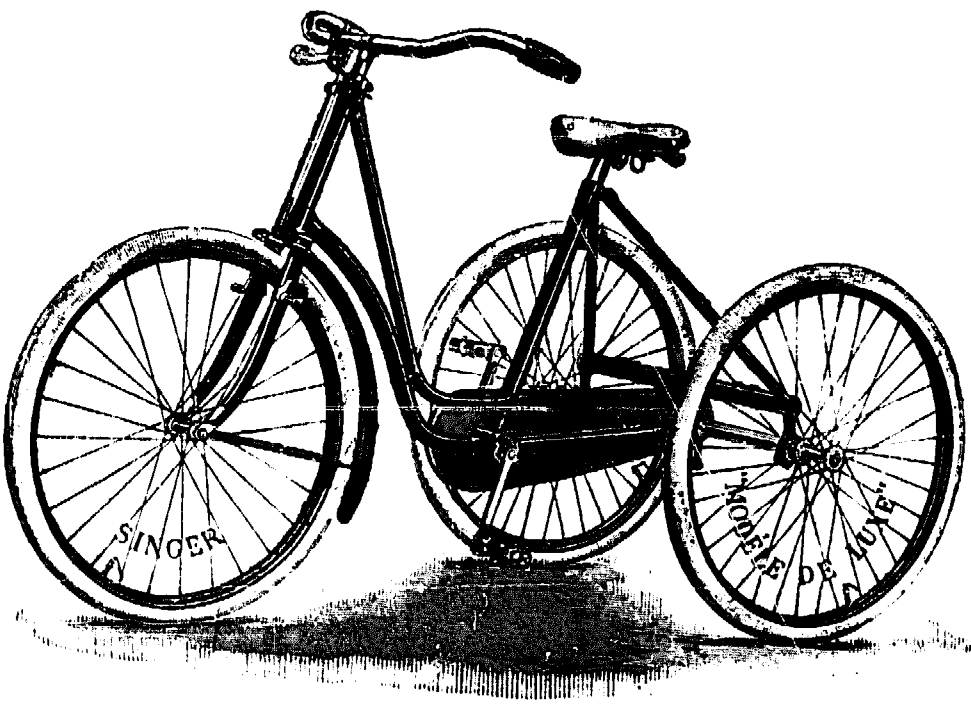


FIG. 273.

in this respect, figure 272 illustrating the frame of the 'Ivel' tricycle. Figure 152 illustrates a tricycle with diamond-frame made by the Premier Cycle Company (Limited). It will be noticed that the frame is the same as that for a bicycle, with the addition of a bridge and four brackets supporting the axle. The next improvement, as regards the proper bracing of the frame, is the spreading of the seat-struts, so that they run towards the ends of the bridge, the bending stresses on the axle-bridge being slightly reduced by this arrangement. Figure 273 shows a tricycle with this arrangement, by Messrs. Singer & Co., but with the front part dropped, so that it may be ridden by a lady.

In nearly all modern tricycles the driving-axle has been supported by four bearings, two near the chain-wheel, so that the pull of the chain can be resisted as directly as possible, and two

at the outer ends, as close to the driving-wheels as possible, each bearing being held in a bracket from the bridge. The whole arrangement of driving-axle, bridge, and brackets looks rather complex, while the chain-struts are subjected to the same severe bending stresses as those of a bicycle (sec. 238). A great improvement is Starley's combined bridge and axle, the bridge being a tube concentric with, and outside, the axle. Figure 274 is

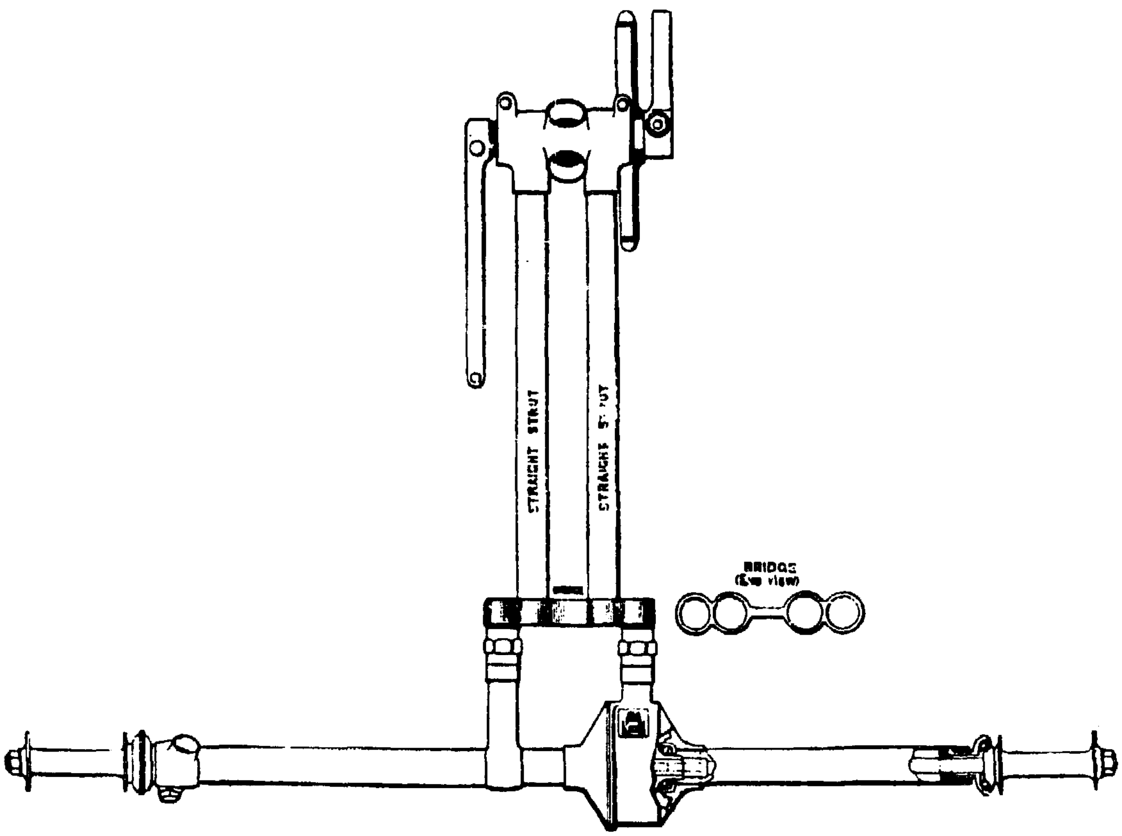


FIG. 274.

a plan showing the arrangement of the combined bridge and axle, crank-bracket and chain-struts, as made by the Abingdon Company, the lug for the seat-strut being shown at the left-hand side of the figure. The driving cog-wheel on the axle is inclosed in an enlarged portion of the outer tube, in which two spaces are made to allow the chain to pass out and in. The chain adjustment is got by lengthening or shortening the chain-struts by means of a right- and left-handed screw, the hexagonal tubular nuts being clearly shown in the figure, an arrangement patented by the author in 1889. Messrs. Starley Brothers have still further improved the tricycle frame by making the chain gear exactly central, so that the design of the frame is simplified by using only one

tube as a chain-strut, while the bending stresses caused by the pull of the chain are eliminated. The crank-bracket (fig. 153) is enlarged at the middle to form a box encircling a chain-wheel, two openings being provided for the chain to pass in and out, as in the axle-box, while three lugs are made on the outside of the box to take the three frame tubes. The narrowest possible tread is thus obtained. This, in the author's opinion, marks the highest level attained in the design of frame for a double-rear-driving tricycle.

224. **Spring-frames.**—In the days of solid tyres many attempts were made to support the frame of a bicycle or tricycle on springs,

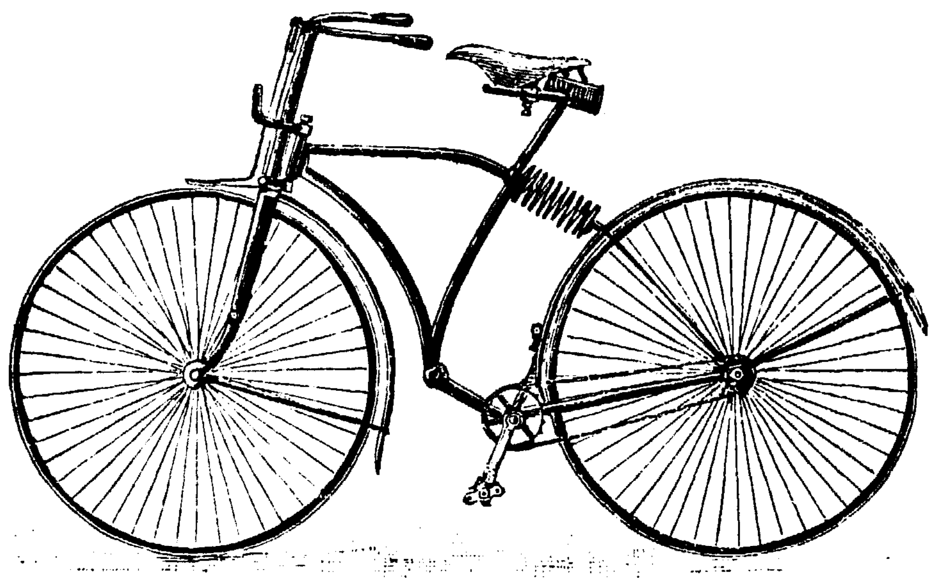


FIG. 275.

so that joltings due to the inequalities of the road might not be transmitted to the frame. The universal adoption of pneumatic tyres has led to the almost total abandonment of spring-frames. The springs should be so disposed that the distances between saddle, handle-bar, and crank-axle remain unaltered. In Harrington's vibration check, which was typical of a number of appliances that could be fitted to the non-driving wheel of a bicycle, the wheel spindle was not fixed direct to the fork ends, but to a pair of short arms fastened to the fork ends and controlled by springs. This allowed the front wheel to move over an obstruction without communicating all the vertical motion to the frame.

Figure 275 shows the 'British Star' spring-frame Safety, made

by Messrs. Guest & Barrow, the rear wheel being isolated by a powerful spring from the part of the frame carrying the saddle.

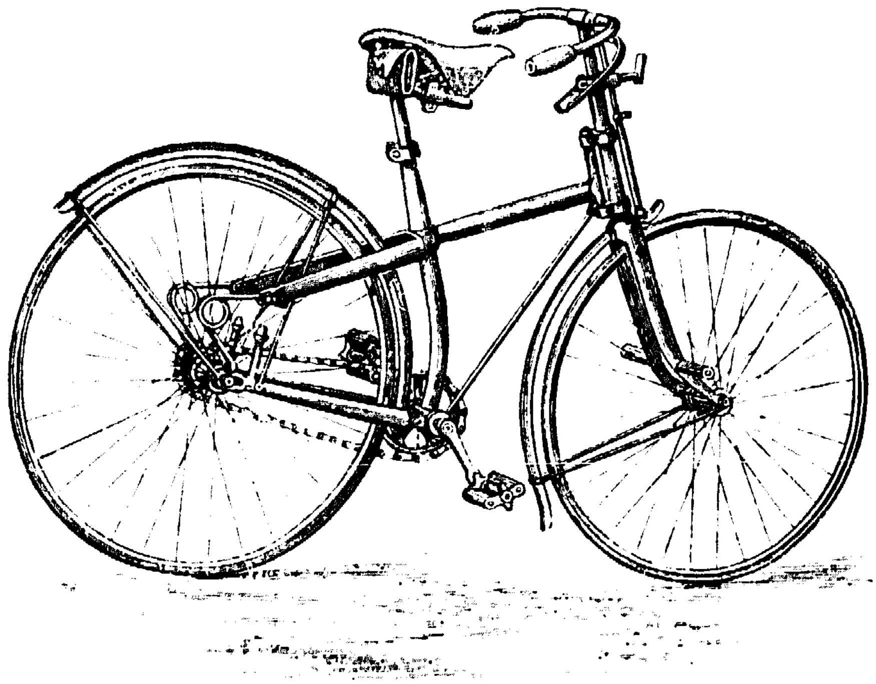


FIG. 276.

Figure 276 shows the 'Cremorne' spring-frame Safety, the springs being introduced near the spindle of the driving-wheel. In

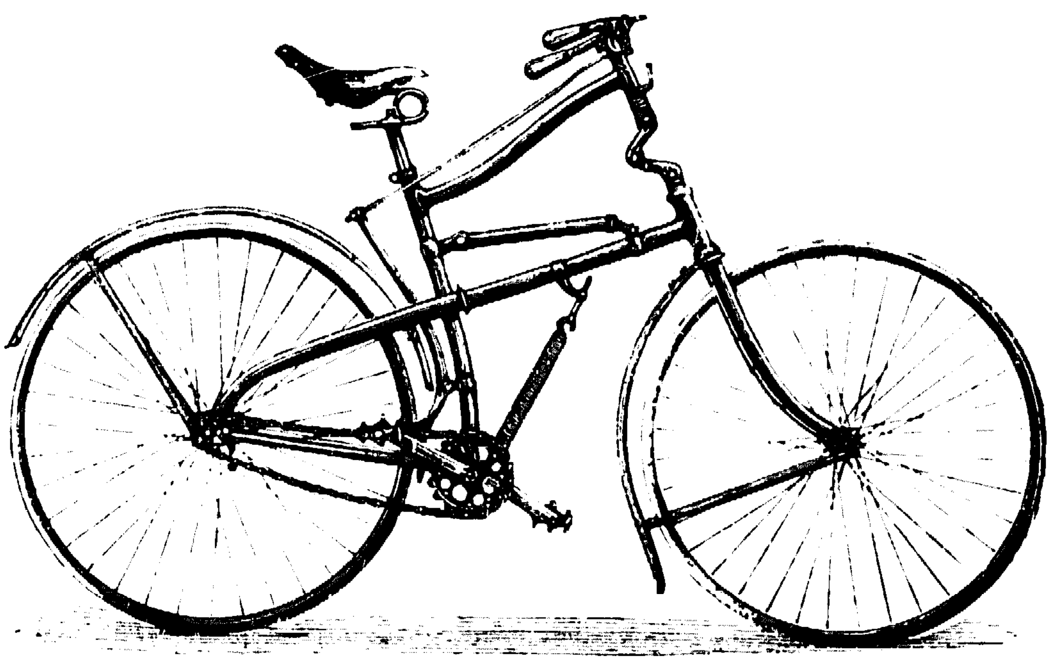


FIG. 277.

both these spring-frames the lower fork is jointed to the frame at or near the crank-bracket. In the 'Elland' spring-frame, made

by Cooper, Kitchen & Co., the spring was introduced just below the seat lug, and the lower fork was hinged to the crank-bracket.

Figure 277 shows the 'Whippet' spring-frame bicycle, the most popular of the type, in which the driving-wheel and steering-wheel forks are carried in a rigid frame. The portion of the frame carrying saddle, crank-axle, and handle-bar is suspended from the main frame by a powerful spiral spring and a system of jointed bars, the arrangements of which are shown clearly in the drawing.

Figure 278 shows a spring-frame bicycle now made by Messrs. Humber & Co. (Limited), the rear fork being jointed to the frame

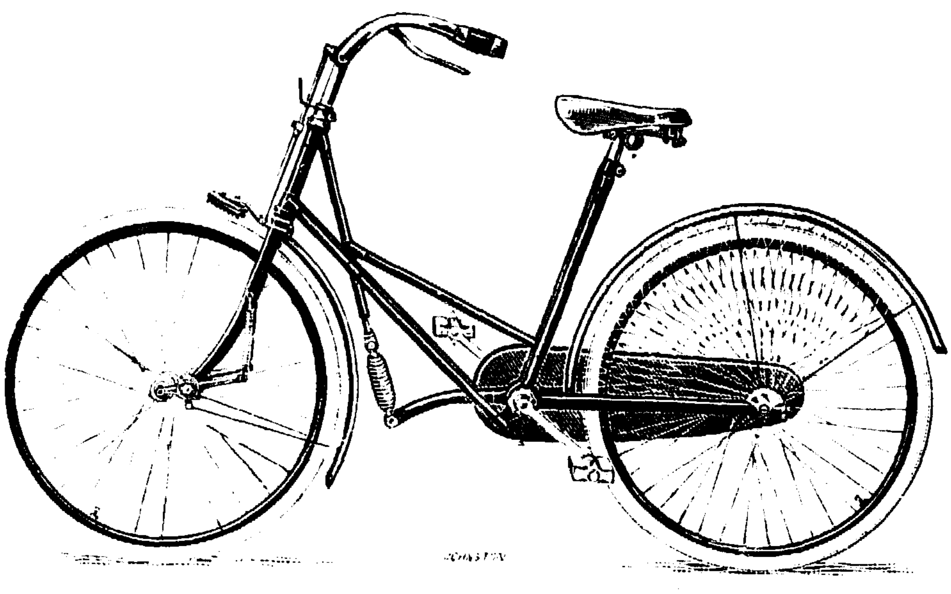


FIG. 278.

at the crank-bracket, and the front wheel being suspended by a pair of anti-vibrators. The rear fork is subjected to a considerable bending moment, and must therefore be made heavy; in this respect the design is inferior to many of the earlier spring-frames.

225. **The Front-frame.**—The front-frames of bicycles and tricycles show great uniformity in general design, any differences between those of different makers being in the details. The front-frame consists of the fork sides, which are now usually tubes of oval section tapered towards the wheel centre; the fork crown; the steering-tube; and the handle-bar. The double-plate fork

crown (fig. 279) is now almost universally used. The fork sides are brazed to the crown-plates. In the best work it is usual to insert a liner at the foot of the steering-tube, shown projecting in figure 279, so as to strengthen the part. The fork tubes are again

FIG. 279.

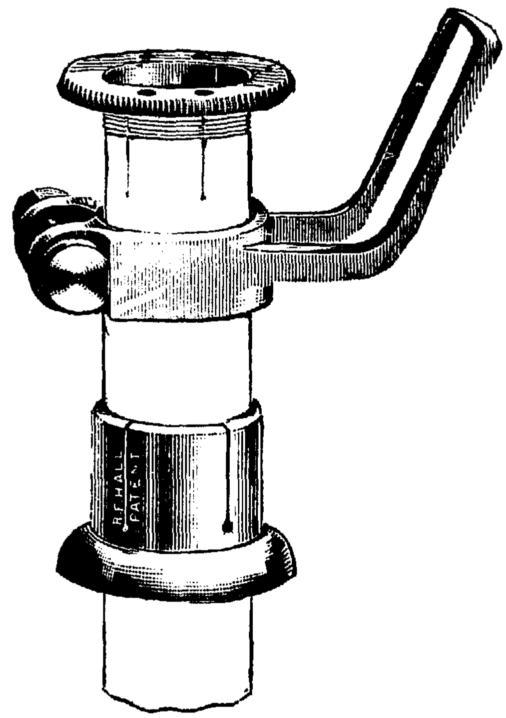


FIG. 280.

strengthened by a liner, the top of which also forms a convenient finish for the fork crown.

The top adjustment cone (fig. 280) of the ball-head is slipped on near the top of the steering-tube, the latter having been previously placed in position through the ball-head. The end of the tube is screwed, to provide the necessary adjustment of the cone. The end of the tube and the tubular portion of the adjustment cone are slit, and the handle-bar having been fixed in the necessary position, the three are clamped together by a split ring and tightening screw. The lamp-bracket is often made a projection from this tightening ring, as shown in figure 280. Figure 280 illustrates the ball-head made by the Cycle Components Manufacturing Company (Limited), and shows the adjustment cone, lamp-bracket, and the adjusting nut apart on the steering-tube, while figure 281 shows the ball-head complete, with the parts assembled in position.

The steering-head of the 'Falcon' bicycle, made by the Yost Manufacturing Company, Toledo, U.S.A., differs from that by the Cycle Components Company, in having the adjusting cone screwed on the steering-tube. The top bearing cup is butted

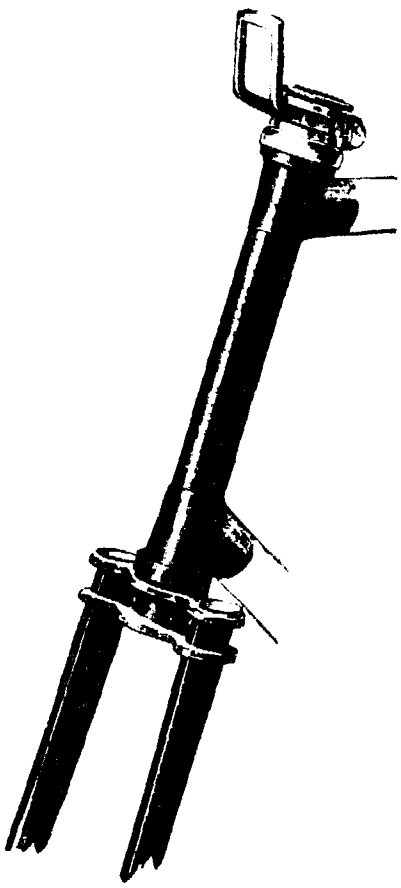


FIG. 281.

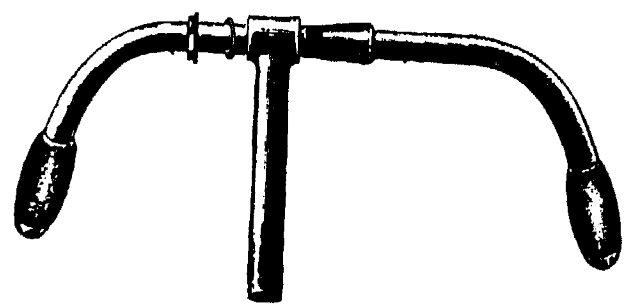


FIG. 282.

against the frame tube of the steering-head, the top lug embracing, and being brazed to, both.

It is becoming more usual not only to make the handle-pillar adjustable in the steering-tube, but also to make the handle-bar adjustable in the socket at the head of the handle-pillar. One of the best designs for accomplishing this (fig. 282) is that used in the 'Dayton' bicycles, made by the Davis Sewing Machine Company, Dayton, U.S.A A conical surface is formed on the handle-bar, and fits a corresponding surface on the socket at the top of the handle-pillar. A short portion of the handle-bar is screwed on the exterior; the handle-bar is fixed in the required position by screwing up a thin nut, and thus wedging the two conical surfaces together.

The handle-bar is most severely stressed at its junction with the handle-pillar. A handle-bar liner (fig. 283), as made by the

FIG. 283.

Cycle Components Manufacturing Company, is used to strengthen it.

The front-frame of the usual type of the present day is essentially a beam subjected to bending, showing in this respect no improvement on that of the earliest tricycles. In tandems and triplets many accidents have resulted from the collapse of the front frame ; additional strength is therefore desirable for this, generally the weakest part of these machines. This can be attained by making the steering-tube and fork sides of sufficient section, and also by entirely new designs for the front-frame.

FIG. 284.

The '*Referee*' front-frame (fig. 284) is made by continuing the fork sides up through the crown to the top of, and outside, the steering-head. The maximum bending-moment is thus resisted by the two fork sides and the steering-tube, instead of only by the latter, as in the ordinary pattern. There should be no possibility, therefore, of the steering-tube giving way.

Duplex fork sides (fig. 285) continued to the top of the steering-head are a still further improvement in the same direction ; the forward tube acting as a strut, the rear tube as a tie, though both are subjected, in some degree, to direct bending.

A braced front frame has been made in the 'Furore' tandem.

In a bicycle designed by the author in 1888, with the object of eliminating, as far as possible, all bending stresses on the frame tubes, the steering-head was behind the steering-wheel, and

consequently the latter could be supported by a trussed frame. The complete frame (fig. 286) had a general resemblance to a

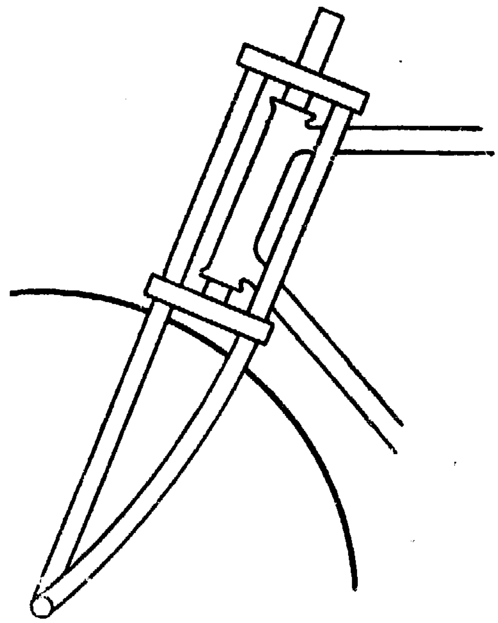


FIG. 285.

queen-post roof-truss. This design answered all requirements as regards lightness and strength; but as an expert rider experienced

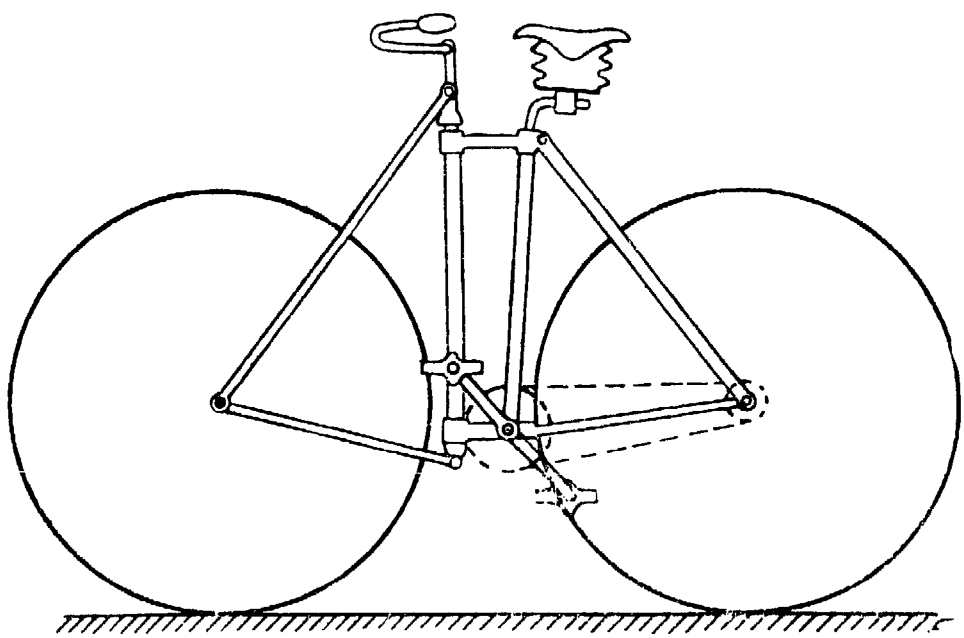


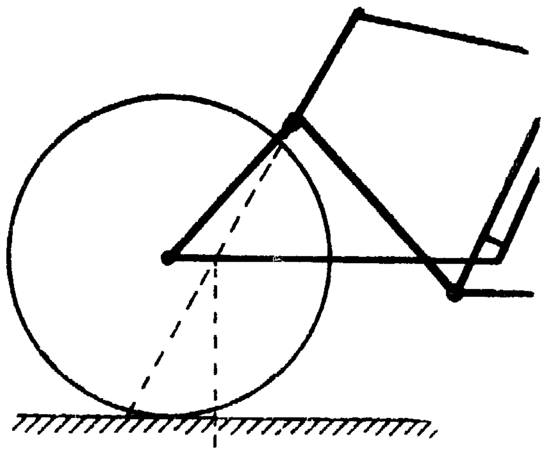
FIG. 286.

almost as much difficulty in learning to ride this machine as a novice in learning to ride one of the usual type, it was abandoned.

In tandems steered by the rear rider, the front-frame could be immensely strengthened by taking stay-tubes from the ends of

the front wheel spindle to a double-armed lever near the bottom of the steering-pillar (fig. 287). These stay-tubes would have to be bent, as shown in plan (fig. 288), to clear the steering-wheel

FIG. 287.



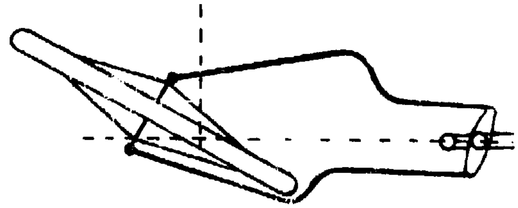


FIG. 288.

when turning a corner. The front fork would then be made straight, as it would act as a strut, while the stress would be almost entirely removed from the steering-tube.

CHAPTER XXIII

THE FRAME (STRESSES)

226. **Frames of Front-drivers.**—$a\ b\ c$ (fig. 290) shows the bending-moment curve on the frame of an 'Ordinary' (fig. 289) due to the weight, W, of the rider. The weight of the rider does not come on the backbone at one point, but, by the arrangement of the saddle spring, at two points, p_1 and p_2. If perpendiculars be drawn from p_1 and p_2 to meet this curve at d_1 and d_2, $d_1\ b\ d_2$ will be the bending-moment curve of the spring, and the remainder $a\ d_1\ d_2\ c$ of the original bending-moment curve will give the bending-moment on the backbone and rear fork. The bending-moment on the backbone is greatest near the head, and diminishes towards the lower end. Accordingly, the backbones of 'Ordinaries' were invariably tapered.

FIG. 289.

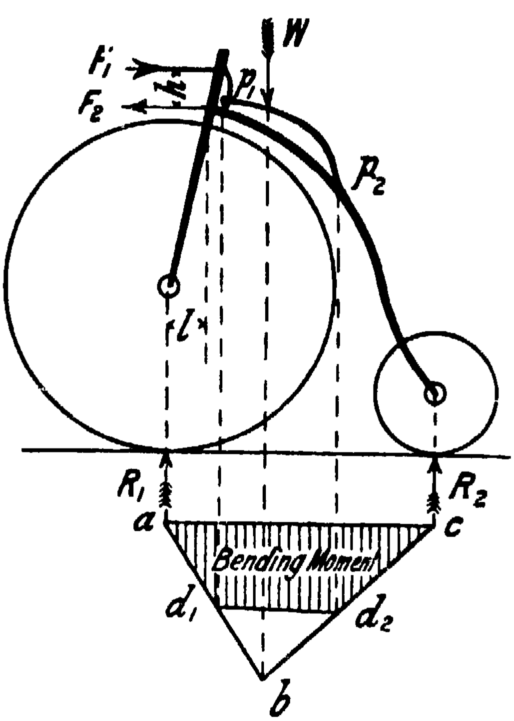


FIG. 290.

In the 'Ordinary' the front fork was vertical, and consequently the bending-moment on the frame just at the steering-head was zero. In the 'Rational,' however, the front fork was sloped, and a bending-moment, $R_1\ l$, existed at the steering-head, l being the horizontal distance of the steering-head behind the front wheel centre. There would consequently be two equal

forces, F_1 and F_2, acting at right angles to the head, at the top and bottom centres, such that

$$F_1 h = R_1 l \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where h is the distance between the top and bottom centres. The greater the distance h, the smaller would be the force F_1, and thus a long head might be expected to work more smoothly and easily than a short one.

It is easily seen that the side pressure on the steering-head of a 'Safety' bicycle or 'Cripper' tricycle arises in exactly the same way. The arrangement of the frame of a 'Safety' is such as permits of a much longer steering-head than can be used in the 'Ordinary,' and as the pressure on the front wheel is much less than in the 'Ordinary,' the side pressure on the steering-head is also very much smaller.

Example I. In a 'Geared Ordinary' the rake of the front fork is 4 inches, the distance between the top and bottom rows of balls in the head is 3 inches, the weight of the rider is 150 lbs., and the saddle is so placed that two-thirds of the weight rest on the front wheel; find the side pressure on the ball-head. The reaction, R_1 (fig. 289), in this case is

$$\frac{2}{3} \times 150 = 100 \text{ lbs.},$$

the bending-moment at the head is

$$100 \times 4 = 400 \text{ inch-lbs.},$$

the force F is therefore

$$\frac{400}{3} = 133{\cdot}3 \text{ lbs.}$$

Example II.—In a 'Safety' bicycle the ball steering-head is $9\frac{1}{2}$ inches long, the horizontal distance of the middle of the head behind the front wheel centre is 9 inches, the rider weighs 150 lbs., and one-fourth of his weight rests on the front wheel; find the side pressure on the steering-head. In this case, the reaction R_1 is

$$\frac{1}{4} \times 150 = 37{\cdot}5 \text{ lbs.},$$

the bending-moment on the head is

$$37 \cdot 5 \times 9 = 337 \cdot 5 \text{ inch-lbs.},$$

the side pressure on the steering-head is therefore

$$\frac{337 \cdot 5}{9\frac{1}{2}} = 35 \cdot 5 \text{ lbs.}$$

Example III.—In Example I., if the point p_1 (fig. 289) of maximum bending-moment on the backbone be 6 inches behind the front wheel centre, find the necessary section of the backbone. The bending-moment at p_1 will be

$$100 \times 6 = 600 \text{ inch-lbs.}$$

If the maximum tensile stress be taken 15,000 lbs. per sq. in., substituting in the formula $M = Zf$, we get

$$Z = \frac{600}{15{,}000} = \cdot 04 \text{ inch-units.}$$

From Table IV., p. 112, a tube $1\frac{1}{4}$ in. diameter, number 20 W.G. would be sufficient.

The section necessary at any other point of the backbone may be found in the same manner, but where the total weight of the part is small, it is usual to make the section at which the straining action is greatest sufficiently strong, and if the section be kept uniform throughout, all the other parts will have an excess of strength. In the backbone of an 'Ordinary,' the section should not diminish by the tapering so quickly as the bending-moment.

Example IV.—In a front-driver in which the load and the relative position of the wheel centres and seat are as shown in figure 291, the stresses can be easily calculated as follows :

Taking the moment about the centre of the rear wheel, we get $R_1 \times 36 = (120 \times 23) + (30 \times 36)$; therefore

$$R_1 = 106 \cdot 7 \text{ lbs.}, \; R_2 = 43 \cdot 3 \text{ lbs.}$$

The maximum bending-moment (fig. 293) on the frame occurs on the section passing through the seat, and is—

$$M = 43 \cdot 3 \times 23 = 996 \text{ inch-lbs.}$$

If the frame simply consists of a backbone formed by a tube $1\frac{1}{4}$ in. diameter, 18 W.G., we find from Table IV., p. 112—

$$Z = \cdot 0525, \text{ and } f = \frac{M}{Z} = \frac{996}{\cdot 0525} = 19{,}000 \text{ lbs. per sq. in.}$$

Braced Frame for Front-driver.—A simple form of braced frame is shown diagrammatically in figure 291. The short tube

FIG. 291.

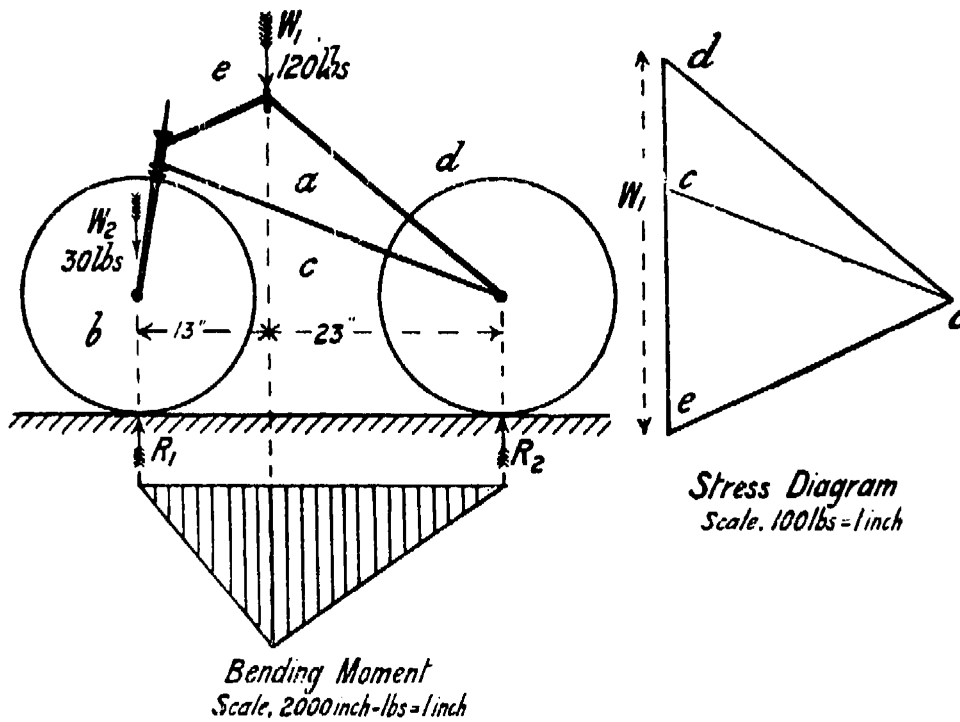


FIG. 293. FIG. 292.

from the steering-head to the seat-lug is made stout enough to resist the bending-moment due to the saddle adjustment, while the seat-struts are subjected to pure compression, and the lower stays to pure tension. Figure 292 shows the stress-diagram for this braced structure; from which the thrust on the seat-struts, *a d*, is 116 lbs. If they are made of two tubes $\frac{5}{8}$ in. diameter, 28 W.G., from Table IV., p. 112, $A = 2 \times \cdot 0284 = \cdot 057$, and the compressive stress is

$$f = \frac{116}{\cdot 057} = 2{,}000 \text{ lbs. per sq. in.}$$

The pull on the lower stay, *a c*, is 95 lbs., and if the stays are made of tubes of the same diameter and thickness as the seat-struts, the tensile stress will be correspondingly low.

The greatest stress on the top-tube will be due to the saddle adjustment. With the horizontal branch of the L pin 3 in. long, a total horizontal adjustment of 6 in. can be provided; the maximum bending-moment on the tube will be

$$M = 150 \times 3 = 450 \text{ inch-lbs.}$$

If the tube be 1 in. diameter, 20 W.G., $Z = \cdot 0253$, and the maximum stress on the tube will be

$$f = \frac{M}{Z} = \frac{450}{\cdot 0253} = 17{,}800 \text{ lbs. per sq. in.}$$

227. **Rear-driving Safety Frame.**—The bending-moment curve for the frame (taken as a whole) of any bicycle is independent of the shape of the frame, and depends on the weight to be carried, and the position of the mass-centre relative to the centres of the wheels. The actual stresses on the individual members of the frame, however, depend on the shape of the frame. The frame of a rear-driving chain-driven Safety must provide supports for the wheel spindle at W, the crank-axle at C, the saddle at S, and the steering-head at H_1 and H_2 (fig. 296). Two principal types of frame are to be distinguished. In the *cross-frame* the point H_1 and H_2 were very close together, and the opposite corners of the quadrilateral $W\,C\,H\,S$ were united by tubes. In the *diamond-frame*, adjacent corners of the pentagon, $H_1\,H_2\,C\,W\,S$, are united by tubes. In both the diamond- and the cross-frames *additional* ties and struts are inserted, the object being to make the frame as rigid as possible, and, of course, to reduce its weight to the lowest possible consistent with strength The weight of a bar necessary to resist a given straining action depends on the magnitude of the straining action and its direction in relation to the bar. We have already seen that a force applied transversely to a bar and causing bending, to be effectually resisted, will require a bar of much greater sectional area than if the force be either direct compression or tension. It may thus be laid down as a guiding principle in designing cycle frames, that *the various members should be so disposed, that as far as possible they are all subjected to direct compression or tension, but not bending*, It follows from this that each member should be attached to

other members at only two points. A bar on which forces can only be applied at two points—its ends—cannot possibly be subjected to bending. If a third 'support' be added, the possibility and probability of subjecting the bar to bending arises. The early Safety frames and some Tandem frames of the present day show many examples of bad design, a long tube often being 'supported' at one or more intermediate points, the result being to throw a transverse strain on it, and therefore weaken, instead of strengthen the structure.

228. **The Ideal Braced Safety Frame.**—In a Safety rear-frame the external forces act on five points; the weight of the

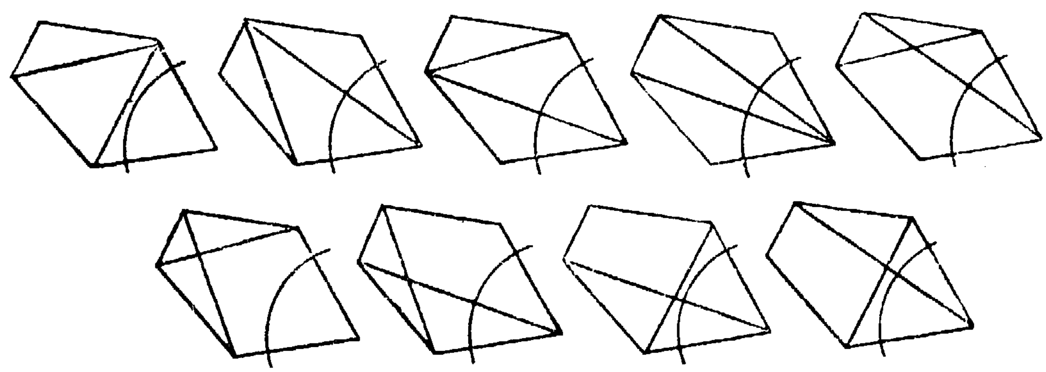


FIG. 294.

rider being applied partially at the saddle S, and at the crank-axle C, the reaction of the back wheel at W, and the pressure on the steering-head at H_1 and H_2 (fig. 296). If the five points H_1, H_2, S, W, and C be joined by bars or tubes dividing the space into triangles, the frame will be perfectly braced, and there will be only direct tensile or compressive stresses on the bars. Figure 296 shows the arrangement used in the 'Girder' Safety frame, while figure 294 shows a number of possible arrangements of perfectly braced rear-frames.

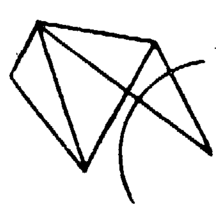


FIG. 295.

Figure 295 shows another perfectly braced rear-frame, in which the lower back fork between the crank-axle and driving wheel spindle is omitted. Comparing this with figure 320, it will be seen that a very narrow tread may be obtained with this frame, a saving of at least the diameter of one lower fork tube being effected.

Example.—The rider weighs 150 lbs., 30 lbs. of which is applied at the crank-axle, the remainder, 120 lbs., at the saddle S.

From the given dimensions of the machine (fig. 296), the reaction, R_1 and R_2, on the front and back wheels can be calculated. Considering the complete frame, and with the dimensions marked, taking moments about the centre of the front wheel, we have

$$(30 \times 23) + (120 \times 33) = R_2 \times 42$$

from which

$$R_2 = 110 \cdot 7 \text{ lbs.}$$

and

$$R_1 = 150 - 110 \cdot 7 = 39 \cdot 3 \text{ lbs.}$$

Consider now the front-frame consisting of the fork and steering-tube; it is acted on by three forces, the reaction, R_1, upwards, and

FIG. 296.

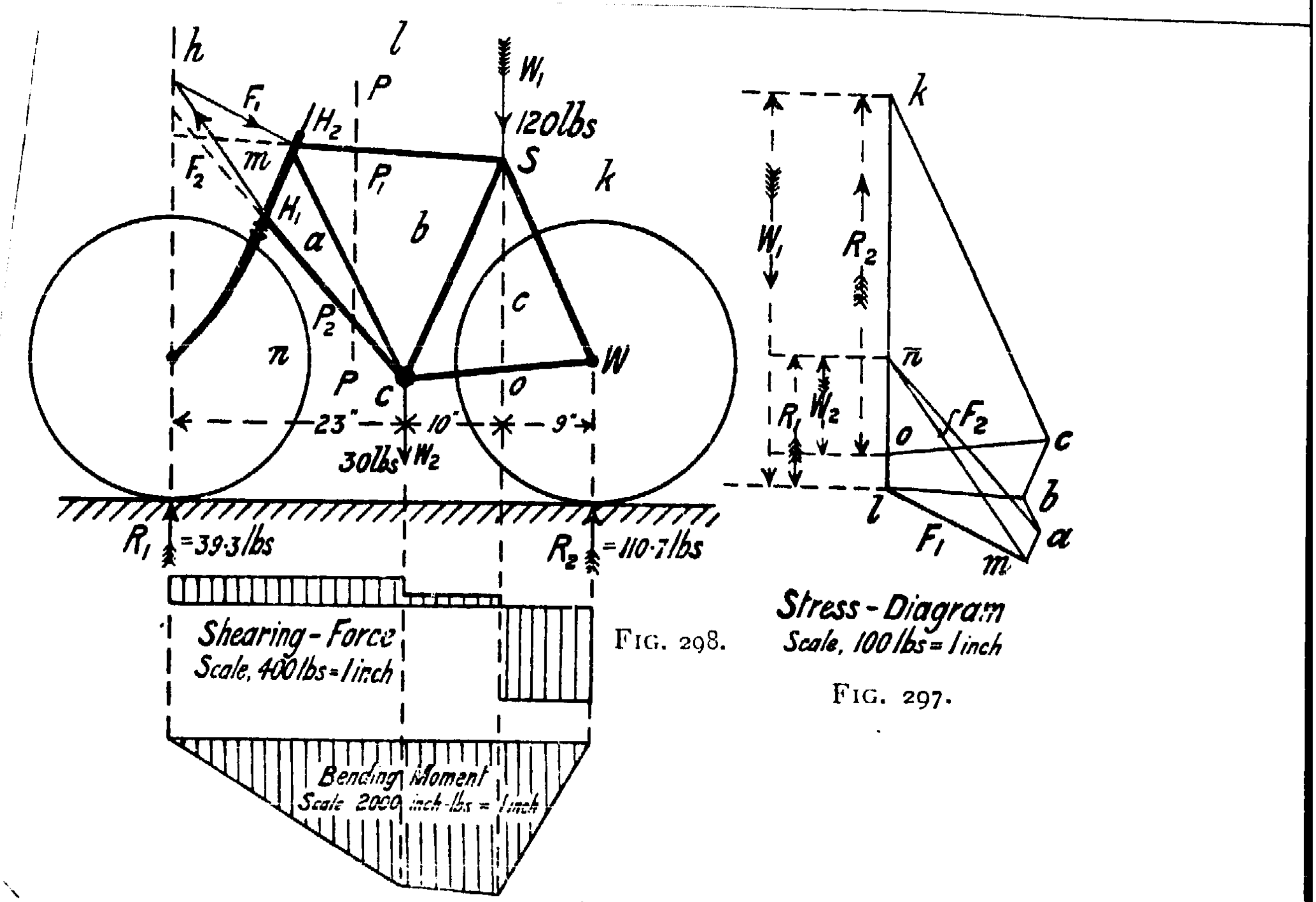


FIG. 298.

FIG. 297.

FIG. 299.

the reactions between it and the rear-frame, at H_1 and H_2. These three forces must therefore (see sec. 45) pass through the same point. With the ordinary arrangement of ball-head the vertical pressure of the front portion of the frame acts on the rear

portion at H_1, and the resultant force at H_2 may be assumed at right-angles to the steering-head, $H_1 H_2$. Therefore, from H_2 draw $H_2 h$, intersecting the vertical through the front wheel centre at h; join $H_1 h$, giving the direction of the force at H_1.

The stress-diagram (fig. 297) can now be constructed by the method of section 83. From this diagram it is easily seen that the lower back fork is in tension, and also the tube from the lower end of the steering-head; the other members of the frame are in compression. By measurement from figure 297 the thrust $k c$ along the seat-struts is 117 lbs.; that along the down-tube, $b c$, about 19 lbs.; along the top-tube, $b l$, about 41 lbs.; along the steering-head, $a m$, 12 lbs.; along the diagonal, $a b$, from the top of the steering-head to the crank-bracket, 7 lbs.; and the tension on the lower back fork, $c o$, is 58 lbs.; on the bottom-tube, $a n$, 64 lbs. These values will, of course, vary slightly according to the dimensions of the frame.

Taking a working stress of 10,000 lbs. per square inch, the sectional area of the two tubes constituting the seat-struts would only require to be $\frac{117}{10000} = \cdot 0117$ sq. in., provided the diameter was great enough to resist buckling. The section of the other tubes would be correspondingly small. We shall see later, however, that many of the frame tubes are subjected to bending, and that the maximum stresses due to such bending are much greater than those considered above.

229. **Humber Diamond-frame.**—The force on the tube between the spaces a and b (fig. 296) is very small, and by careful design may be made zero (see sec. 230). In the 'Humber' diamond-frame this tube is suppressed, and thus if the frame tubes were connected by pin-joints at C, S, H_1 and H_2, the frame would be no longer able to retain its form when subjected to the applied forces. That the frame actually retains its shape is due to the fact that the frame joints are rigid, and that the individual members are capable of resistance to bending. If all the frame joints are rigid, the stress in any member cannot be determined by statical methods, but the elasticity and deformation of the parts under stress must be considered. However, by making certain assumptions, results which may be approximately true can be obtained by statical methods.

Example I.—Suppose the tubes $C\ H_1$ and $H_1\ H_2$ to be fastened together at H_1, so as to form one rigid structure, which we may consider connected by pin-joints to the frame at C and H_2, the other joints of the frame being pin-joints. The distance of H_1 from the axis of the suppressed member, $H_2\ C$, is 6 in. The bending-moment at H_1 on the part $C\ H_1\ H_2$ is therefore 7 lbs. × 6 in. = 42 inch-lbs.; the bending-moment diminishes towards zero at C and H. If the tube $H_1\ C$ be 1 in. diameter, and a working stress of 10,000 lbs. per sq. in. be allowed, substituting in formula (3), section 101, we get

$$10{,}000 = \frac{64}{A} + \frac{4 \times 42}{A \times 1}$$

or,

$$A = \cdot 0232 \text{ sq. in.}$$

Consulting Table IV., p. 112, we see that the thinnest there given, No. 32 W.G., has an excess of strength. If the tube $H_2\ C$ had been retained, the sectional area of the tube $H_1\ C$ need only have been

$$A = \frac{64}{10{,}000} = \cdot 0064 \text{ sq. in.}$$

Example II.—Suppose the tubes $H_1\ H_2$ and $H_2\ S$ rigidly fastened at H_2, and connected at H_1 and S by a pin-joint to the rest of the frame. The part $H_1\ H_2\ S$ may then be considered as a beam carrying a load of 7 lbs. at H_2. The perpendicular distances of H_1 and S from the line of action of this force are 6 in. and 18 in. respectively. The bending-moment at H_2 is therefore (see sec. 87)

$$\frac{7 \times 6 \times 18}{24} = 31 \cdot 5 \text{ inch-lbs.}$$

The compressions along $H_1\ H_2$ and $H_2\ S$ will be increased by the components of the original force, 7 lbs., along the suppressed bar at $H_2\ C$. Similarly, the forces along $C\ H_1$ and $C\ S$ will be altered. The thickness of tube required can be worked out as in Example I. above.

230. **Diamond-frame, with no Bending on the Frame Tubes.**—Consider the complete frame divided by a plane, $P\ P$ (fig. 296) immediately behind the steering-head, $H_1\ H_2$. If the frame

tubes $H_2 S$ and $H_1 C$ are not subjected to bending, the forces exerted by the front part of the frame on the rear part must be in the direction of the tubes. The forces acting at P_1 and P_2 on the front portion of the frame are equal in magnitude but reversed in direction. The only other external force acting on the front portion of the frame is the reaction, R_1, of the wheel on the spindle; these three forces are in equilibrium, and therefore must all pass through the same point. The condition then that the tubes in a diamond-frame should be subjected to no bending is that *the axes of the top- and bottom-tubes should, if produced, intersect at a point vertically over the front wheel centre.* This is very nearly the case in figure 296; if it was exactly, the force $b\ a$ along $H_2 C$ would be zero.

231. **Open Diamond-frame.**—The open diamond-frame (figs. 127, 246), though in external appearance very like the 'Humber' frame, is subjected to totally different straining actions. In the first place, if the joints at C, H_1, H_2, S and W be pin-joints, under the action of the forces the frame would at once collapse. Practically, the top-tube, $H_2 S$, and the seat-struts, $S\ W$, form one rigid beam, which must be strong enough to resist the bending-moment due to the load at S. Taking the same dimensions as in figure 296, the distances of H_2 and W from the line of action of the load at S are 21 in. and 9 in. respectively, and the weight of the rider 150 lbs., the bending-moment at S will be

$$\frac{150 \times 21 \times 9}{30} = 945 \text{ inch-lbs.}$$

Taking $f = 20{,}000$ lbs. per sq. in., and substituting in the formula $M = Zf$,

$$Z = \frac{945}{20{,}000} = \cdot 0472 \text{ in.}^3$$

From Table IV., p. 112, a tube 1 in. diam., 14 W.G., would be required; or a tube $1\frac{1}{8}$ diam., 17 W.G.

When the rider is going easily his whole weight rests on the saddle, and must be supported by the beam $H_2 S\ W$. On the other hand, when working hard, as in riding up a steep hill, his whole weight may be applied to the pedals, and, therefore, will come on the frame at C. The bottom-tube, $H_1\ C$, and the lower

back fork, $C\,W$, must be rigidly jointed together at C, and form a beam sufficiently strong to resist this bending-moment. Taking the same dimensions as in figure 296, the bending-moment at C is

$$\frac{150 \times 15 \times 19}{34} = 1{,}257 \text{ inch-lbs.}$$

$$Z = \frac{M}{f} = \frac{1{,}257}{20{,}000} = {\cdot}0628 \text{ in.}$$

and a 1 in. tube, 11 W.G., or a $1\frac{1}{8}$ in. tube, 14 W.G., would be required. A comparison of these results with those of sections 228-9 will reveal the weakness of the open diamond-frame.

232. **Cross-frame.**—In the cross-frame (fig. 300), the forked backbone a runs straight from the steering-head to the back

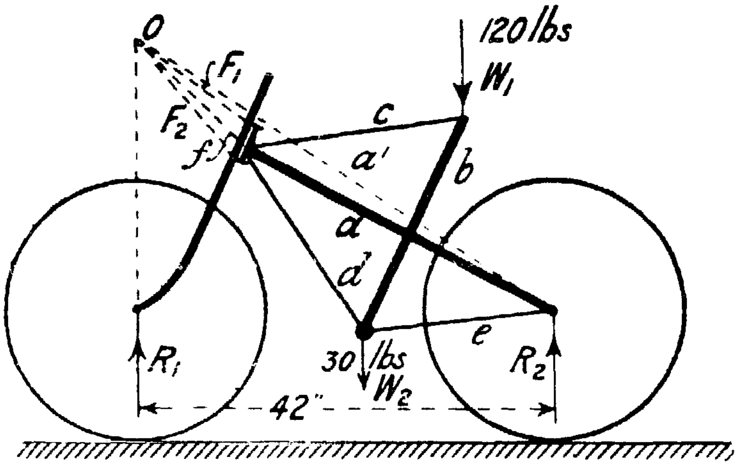


FIG. 300.

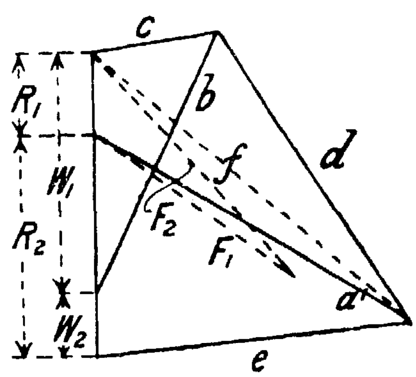


FIG. 301.

wheel spindle. The crank-bracket and seat-lug are connected by the down-tube b. The earlier cross-frames consisted only of these two members; but in the later ones, bottom stays e, from the crank-bracket to the back wheel spindle, and stays c and d, from the steering-head to the crank-bracket and seat-lug respectively, were added. With this arrangement, the down-tube b is subjected to thrust, the stays c, d, and e to tension, and the backbone a to thrust, combined with bending, due to the forces acting on the steering-head.

The stress-diagram can be drawn as follows: The loads W_1 and W_2 at the seat-lug and crank-axle respectively being given, the reactions R_1 and R_2 at the wheel centres can be calculated, as in section 89. At the back wheel spindle three forces act. The

magnitude and direction of R_2 are known, the pull of the lower fork *e* is in the direction of its axis, and is, therefore, known, but the thrust on the backbone *a* is not along its axis ; its direction is not known, and we cannot, therefore, begin the stress-diagram with the forces acting at this point. If the initial tension on the stays *c* and *d* be assumed such that there is no straining action at the junction of the down-tube *b* and the backbone *a*, the former will be subjected to a thrust along its axis, and, therefore, the directions of the three forces acting at the seat-lug are known. Draw W_1 (fig. 301), equal and parallel to the weight acting at the seat-lug, and complete the force-triangle $W_1\ b\ c$, the sides being parallel to the correspondingly lettered members of the frame (fig. 300). Proceeding to the crank-bracket, the forces acting are the weight W_2, the thrust of the down-tube *b*, and the pulls of the stays *d* and *e*, the directions of which are known ; the force-polygon $b\ W_2\ e\ d$ can therefore be drawn. Proceeding now to the back wheel spindle, the pull of the stay *e* and the upward reaction R_2 are known. Setting off R_2 (fig. 301) from the extremity of the side *e*, and joining the other extremities of *e* and R_2, the direction and magnitude a^1 of the thrust on the backbone are obtained. This thrust does not act along the axis of the backbone, which is, therefore, in addition to a thrust along its axis, subjected to a bending-moment varying from zero at the back wheel spindle to a maximum $P\,l$ at the steering centre, P being the thrust measured from figure 301, and l the distance of a perpendicular on the line of action of the thrust from the centre of the backbone at the steering-head.

The forces acting on the backbone *a* are : the pull of the lower stay *e* and the reaction R_2, having the resultant a^1 ; the pulls *c* and *d*, with resultant *f* acting, of course, through the point of intersection of *c* and *d* ; the pressures F_1 and F_2 on the steering centres. Since R_1, F_1 and F_2 reversed are the only forces acting on the front-frame, the resultant of F_1 and F_2 must be R_1. Thus the forces acting on the backbone *a* are equivalent to the three forces R_1, a^1 and *f*, which must, therefore, all pass through the point *O*. A check on the accuracy of the stress-diagram is thus obtained.

The point *O* being determined, by joining it to the top and

bottom steering centres the directions of F_1 and F_2 are obtained, and their magnitudes by drawing the force-triangle $R_1 F_1 F_2$ (fig. 301).

233. **Frame of Ladies' Safety.**—Figure 302 is the frame-diagram of a Ladies' Safety. Having given the loads at the seat-lug and crank-axle, the reaction of the wheels can be calculated, as in section 89. The stresses on the seat-struts *a* and the back fork *e* may be found in exactly the same manner as for the diamond frame; viz. by drawing the force-triangle $R_2\,a\,e$ (fig. 303). The best arrangement of the two tubes *c* and *d* from the steering-head

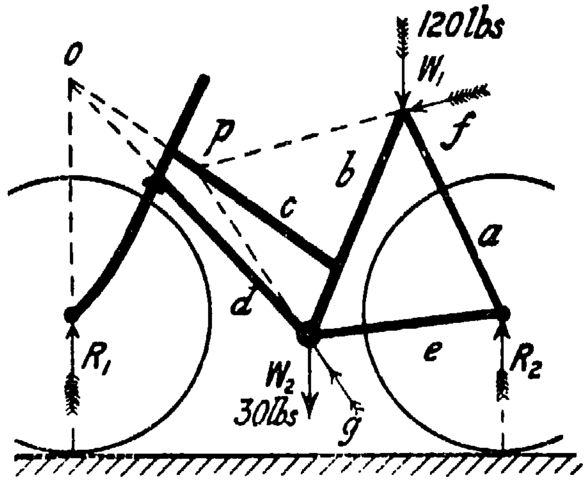


FIG. 302.

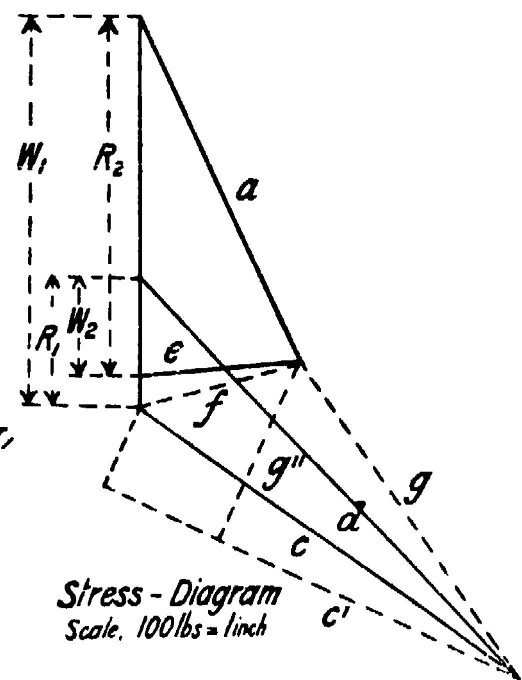


FIG. 303.

will be when their axes intersect at a point *o* vertically over the front wheel centre. Assuming that the forces on these two tubes are parallel to their axes, they are determined by drawing the force-triangle $R_1\,d\,c$ for the three forces (fig. 303) acting on the front-frame of the machine. The down-tube *b* is acted on by three forces—(1) The thrust of the tube *c*; (2) the resultant *f* of the thrust *a* and load W_1 acting at the seat-lug; (3) the resultant *g* of the pulls *d* and *e* and the load W_2, acting at the crank-axle. These three forces form the force-triangle *c f g* (fig. 303). A check on the accuracy of the work is obtained by

the fact that the forces f and g (fig. 302) must intersect at a point p on the axis of the tube c.

From figure 303, the thrust of c on the down-tube b is 145 lbs., while its component c^1 at right angles to b is 142 lbs. The down-tube b is 22 in. long, divided by c into two segments, 7 and 15 in. The greatest bending-moment on it is therefore

$$\frac{142 \times 7 \times 15}{22} = 667 \text{ inch-lbs.}$$

The lower part of the down-tube is subjected to a thrust g'' (the component of the force g parallel to the down-tube) of 62 lbs.

234. **Curved Tubes.**—About the years 1890–2 a great number of Safety frames were made with the individual tubes curved in various ways. The curving of the tubes was made on æsthetic grounds, and possibly the tremendous increase in the maximum stress due to this curving was not appreciated. The maximum stress on a curved tube subjected to compression or tension at its ends is discussed in section 101.

Example.—Let a tube be bent so that its middle point is a distance equal to four diameters from the straight line joining its ends; the maximum stress is, by (4) section 101,

$$\frac{P}{A} + \frac{4 \times P \times 4d}{Ad} = \frac{17P}{A}$$

The tube would, therefore, have to be seventeen times the sectional area of a straight tube subjected to the same thrust.

235. **Influence of Saddle Adjustment.**—So far we have considered the mass-centre of the rider to be vertically over the point S (fig. 296); this is approximately the case when the saddle is fixed direct, as in some racing machines, to the top of the back fork without the intervention of an adjustable saddle-pin. But when an adjustable saddle-pin is used, the weight on the saddle acts at a distance l, usually from 3 to 6 inches behind the point S. The weight W acting at this distance is equivalent to an equal weight acting at S, together with a couple Wl, producing a bending-moment $W_1 l$ at S. From the manner in which the adjustable pillar is usually fixed at S, this bending-moment is generally transmitted to the down-tube SC, which must therefore be stout enough to resist it. Since, however, the

joint at S is rigid, a small part of this bending-moment may be transmitted to the tube $S H_2$.

Example.—Taking $l = 5$ in., and $W = 120$ lbs., as in figure 296, $M = 120 \times 5 = 600$ inch-lbs. Taking the direct thrust along $S C$ 19 lbs., as in section 228, and a working stress 20,000 lbs. per sq. in., the diameter of the tube 1 in., and substituting in (3) section 101,

$$20{,}000 = \frac{19}{A} + \frac{4 \times 600}{A},$$

we find $A = \cdot 1210$ sq. in. From Table IV., p. 113, the tube would require to be 18 W.G.

If the saddle were placed vertically over S, and no bending came on the tube $S C$, its sectional area would be $\frac{19}{20{,}000}$ $= \cdot 00095$ sq. in.: one-hundredth part of the section necessary with the saddle placed sideways from S.

This example is typical of the enormous additions which must be made to the weights of the tubes of a frame when the forces do not act exactly along the axes of the tubes.

By the use of the T-shaped seat-pillar (fig. 304) the range of horizontal adjustment can be increased without increasing unduly

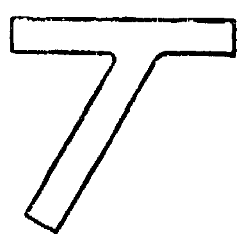


FIG. 304.

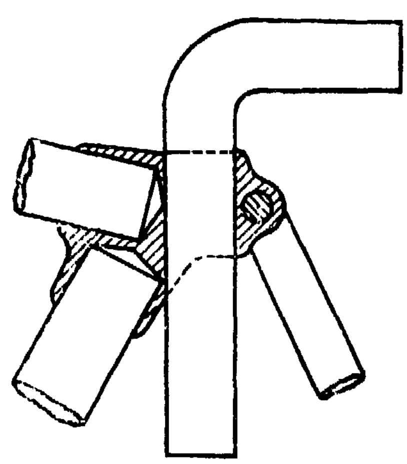


FIG. 305.

the stresses due to bending; or, for a given range of horizontal adjustment, the bending stresses are lower with the **T**-shaped than with the **L**-shaped seat-pillar. The adjustment got by an **L** pin with horizontal and vertical limbs is much better (figs. 256, 260), since by turning the **L** pin round, the saddle may be adjusted either before or behind the seat-lug S. Thus,

for a horizontal adjustment of 6 inches, the maximum eccentricity l need not be greater than 3 inches. By combining such an L pin with the 'Humber' frame it would be possible to further reduce the stresses on the frame. Figure 305 shows a seat-lug for this purpose, designed by the author.

For racing machines of the very lightest type possible the best result is obtained by fastening the saddle direct at S; this, of course, does not allow of any adjustment, and a machine that might suit one rider admirably might not be suitable for others.

236. **Influence of Chain Adjustment.**—In chain-driven Safeties it is found that chains stretch, no matter how carefully made, after being some time in use, and therefore some provision must be made for taking up the slack. This is usually done by making the distance between the centres of the crank-axle and the driving-wheel adjustable. Figure 306 shows a common faulty design for the stamping at the driving-wheel spindle. The force R_2 (fig. 306) is equivalent to an equal force acting at W plus a bending-moment $R_2\,l$, which is transmitted to the upper and lower forks.

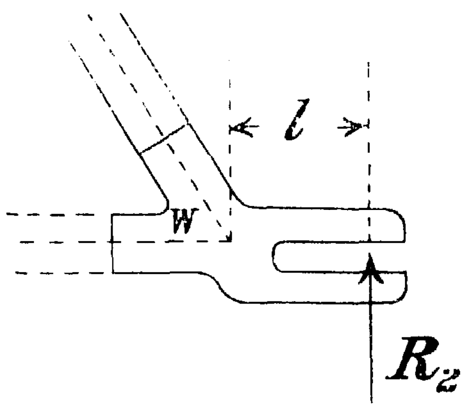


FIG. 306

Example.—If the distance l (fig. 306) be $\frac{1}{2}$ inch, and R_2 be, as in the example of section 228, 111 lbs., the bending-moment transmitted to the forks is 55·5 inch-lbs. The direct compression along the seat-struts $S\,W$ is 41 lbs. (fig. 297), that along the lower fork $W\,C$ is 58 lbs. Taking 10,000 lbs. per sq. in. as the working stress of the material, a section of ·0041 sq. in. for the top fork, and ·0058 sq. in. for the bottom fork would be sufficient, if they were not subjected to bending. Suppose the bending to be taken up entirely by the lower fork, made of two tubes $\frac{3}{4}$ in. diameter, and of total area A; then, when subjected to bending as well as to direct compression or tension, the maximum stress to which they are subjected is given by the formula (3) of section 101. Substituting the above numerical values of f, P, M, and d, we have

$$10{,}000 = \frac{58}{A} + \frac{4 \times 55{\cdot}5}{A \times {\cdot}75},$$

or $A = \cdot 035$ sq. in. Thus the maximum stress, when the force R_2 is applied $\frac{1}{2}$ in. from the point of intersection of the forks, is nearly seven times as great as when it is applied in the best possible position.

Swinging Back Fork.—The centre of the driving-wheel may be always at the intersection of the top and bottom forks if the top fork be attached to the frame at S by a bolt—the bolt used for tightening the saddle-pin may serve for this purpose—and its lower ends be provided with eye-holes for the reception of the spindle of the driving-wheel. This arrangement, now almost universal, was first designed by the author in 1889. The lower fork may then be provided with a plain straight slot (fig. 307), along which the wheel spindle can be pulled by an adjusting screw. During a small adjustment of this nature the angle

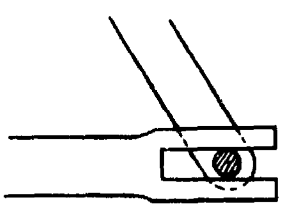


FIG. 307

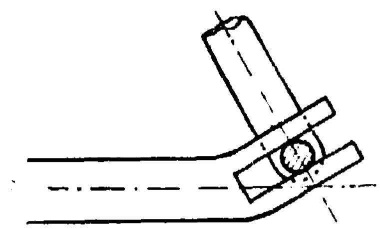


FIG. 308.

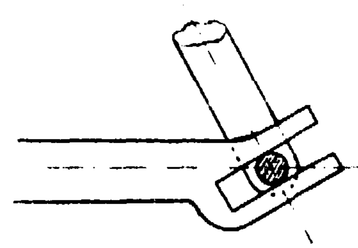


FIG. 309.

$S\,C\,W$ (fig. 296) will vary slightly, so that theoretically the lower forks should be attached by a pin-joint at C; but practically the elasticity of the tubes is sufficiently great to allow of the use of a rigid joint. In the form of this adjustment used by Messrs. Humber & Co. the slot is not in the direction of the axis of the lower fork, but curved (fig. 308) to a circular arc struck from S as centre. In this way there is no tendency to alter the angle $S\,C\,W$ (fig. 296); but the fact that the centre of the wheel spindle does not always lie on the axis of the lower fork $C\,W$ throws a combined tension and bending on it, the bending-moment being equal to $P\,l$, where P is the direct force on the lower fork parallel to its axis, and l is the distance of the centre of the wheel from the axis of the lower fork.

Example.—Let $l = \frac{d}{2}$, that is, the centre of the wheel is just on a line with the top of the tube of the lower fork.

Substituting in (3), section 101,

$$f = \frac{P}{A} + \frac{4\,P\,d}{2\,A\,d} = \frac{3\,P}{A}.$$

If the centre of the wheel lay on the axis of the tube the stress would be uniformly distributed and equal to $\frac{P}{A}$. Thus the stress on the lower fork is increased by the eccentricity of the force acting on it to three times its value with no eccentricity.

A better arrangement for the slot would be that shown in figure 309, where the spindle is adjusted equally above and below the centre line of the lower fork tubes.

237. **Influence of Pedal Pressure.**—In the foregoing discussion we have considered the forces to be applied in the middle plane of the bicycle frame; but the rider applies pressure on the pedals at a considerable distance from the middle plane, and thus additional transverse straining actions are introduced. We now proceed to investigate the corresponding stresses.

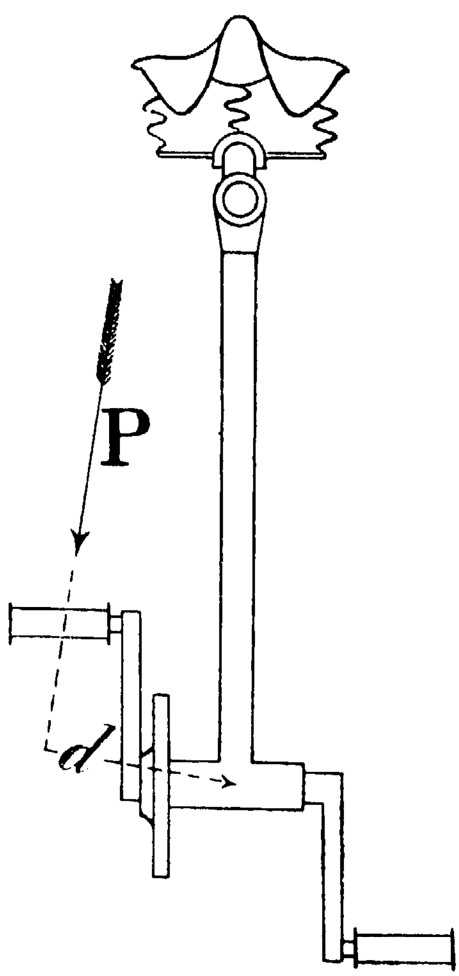


FIG. 310.

Figure 310 is a transverse sectional elevation, showing the pedals, cranks, crank-bracket, saddle, and down-tube, to the foot of which the crank-bracket is fixed. A force, P, applied to the pedal will cause a bending of the crank-bracket, which will be transmitted to the down-tube. From the arrangement of the lower fork in relation to the crank-bracket it is seen that practically none of this bending-moment can be transmitted to the lower fork. A small portion of the bending-moment may be transmitted to the bottom-tube $H_1\,C$ (fig. 296), but the greater part will be transmitted to the down-tube.

The magnitude of the bending-moment is $P\,d$, d being the

length of the perpendicular from the centre of the crank-bracket on to the line of action of the force P. The narrower the tread the smaller will be d, and therefore the smaller the transverse stresses on the frame. Hence the importance of obtaining a narrow tread.

Example I.—Let P be 150 lbs., the tread, *i.e.* the distance from centre to centre of the pedals measured parallel to the crank-axle, 11 inches. The distance d may be taken equal to half the tread, *i.e.* $5\frac{1}{2}$ inches. The bending-moment on the foot of the down-tube will be $150 \times 5\frac{1}{2} = 825$ inch-lbs. Let the down-tube be $1\frac{1}{8}$ in. diameter, 20 W.G. From Table IV., p. 113, the Z for the section is $\cdot 0325$ in.^{3}; substituting these values in the formula $M = Zf$, we get

$$825 = \cdot 0325 f,$$

i.e. $f = 25{,}400$ lbs. per sq. in.

Compared with the result on page 310, got by considering the forces applied in the middle plane of the frame, it is seen that on the down-tube the stress due to transverse bending is the most important.

In the double diamond-frame the single down-tube of figure 310 is replaced by the two tubes which support the crank-bracket near its ends (fig. 311). This gives a much better construction to resist the transverse stresses, but unfortunately it is not so neat in appearance as the single tube, and its use has been practically abandoned of recent years. The maximum stress produced in this case can be easily calculated and may be illustrated by an example.

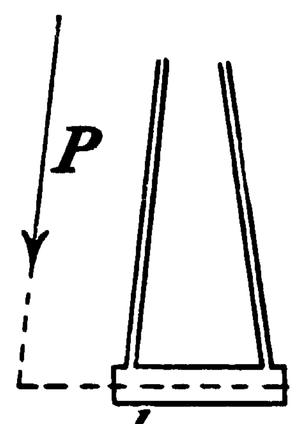


FIG. 311.

Example II.—Let the tubes be $\frac{3}{4}$ in. diameter, 20 W.G., with their ends 3 in. apart. Under the action of the force P the nearer tube will be subjected to tension, the further one to compression. Taking moments about a (fig. 311), the point of attachment of the further tube to the crank-bracket, we get

$$7\,P = 3\,F, \text{ i.e. } F = \frac{7}{3} P = 350 \text{ lbs.}$$

The sectional area of the tube, from Table IV., p. 113, is ·0807 sq. in., therefore the stress on the tube is

$$f = \frac{350}{·0807} = 4,336 \text{ lbs. per sq. in.}$$

238. **Influence of Pull of Chain on Chain-struts.**—In riding easily along a level road, when very little effort is being exerted, the tension on the chain is small, and the stresses on the lower back fork, or chain-struts, will be as discussed in section 228. But when considerable effort is being exerted on the pedal, the tension on the chain is considerable, and since the chain does not lie in the middle plane of the frame, additional straining actions are introduced.

The tension F on the chain (fig. 312) can be easily found by considering the single rigid body formed by the pedal-pins, cranks, crank-axle, and chain-wheel. This rigid body is free to turn about the geometric axis of the crank-bracket, and it is acted on by three forces: P the pressure on the pedal-pin, the pull F of the chain, and the reaction of the balls on the crank-axle. Taking moments about the geometric-axis of the axle, that of the latter force vanishes, and we get $Pl = Fr$; l being the length of crank, and r the radius of the sprocket-wheel.

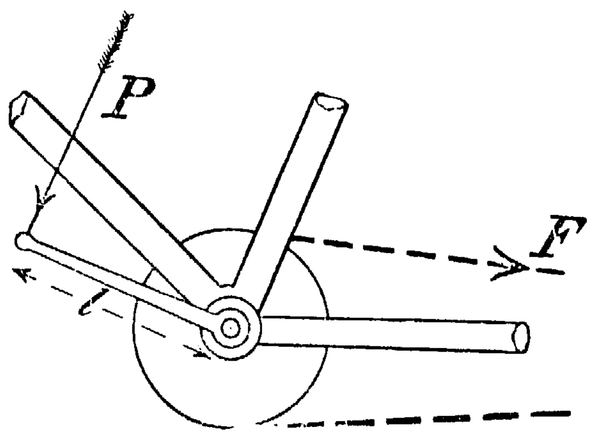


FIG. 312.

Example I.—Let $P = 150$ lbs., $l = 6\frac{1}{2}$ in., and let the chain-wheel have eighteen teeth to fit the 'Humber' pattern chain. From Table XV., p. 405, we get $r = 2·87$ in.; therefore

$$2·87\, F = 6\tfrac{1}{2}\, P, \text{ and } F = \frac{6·5}{2·87} \times 150 = 340 \text{ lbs}$$

Figure 313 is a plan showing the crank-bracket and the lower back fork. Consider the horizontal components of the forces acting on the crank-bracket. If the pressure on the pedals be vertical there will be no horizontal component due to it, and we

are left with the force F_1, the horizontal component of the pull on the chain. This is equilibrated by the horizontal components

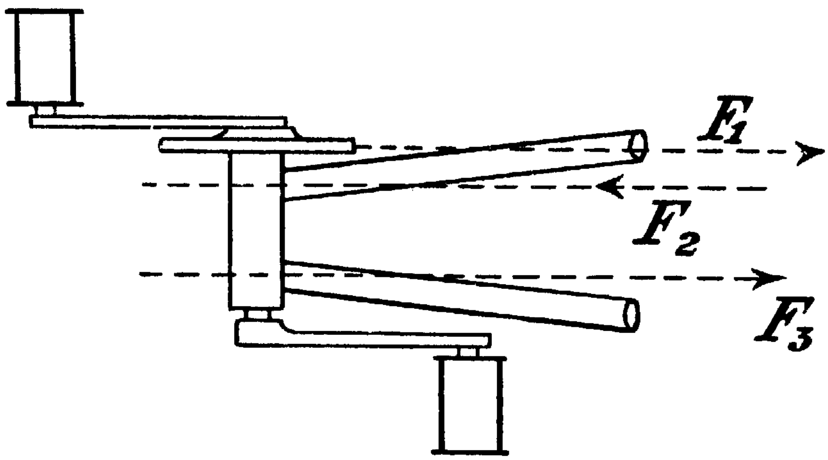


FIG. 313.

of the reactions at the bearings, therefore the crank-bracket i acted on by the forces at the bearings and the forces F_2 and F_3 exerted by the ends of the lower back fork.

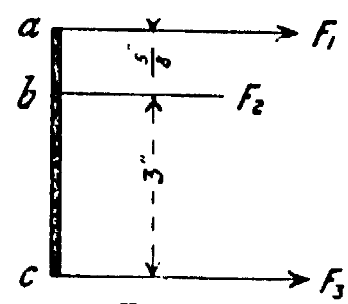


FIG. 314.

Example II.—Let the chain-line be $2\frac{1}{8}$ in. (*i.e.* the distance from the centre of the chain to the centre of the fork is $2\frac{1}{8}$ in.), let the fork ends at the crank-bracket be 3 in. apart; then the forces to be considered are shown in figure 314.

To find the pull F_3, take moments about *b*.

$$\tfrac{5}{8}\,F_1 = 3\,F_3, \text{ therefore } F_3 = \frac{\cdot625}{3}\,340 = 70\cdot8 \text{ lbs.}$$

To find the compression F_2 on the near tube, take moments about *c*, and we get $3\frac{5}{8}\,F = 3F_2$,

$$\therefore F_2 = \frac{3\cdot625}{3}\,340 = 410 \text{ lbs.}$$

Comparing with the results of section 228, the compression on the near tube of the fork is much greater than the tension due to the weight of the rider applied centrally. The near tube, therefore, must be designed to resist compression.

Bending of Chain-struts.—The sides of the lower back

fork, the crank-bracket, and the back wheel spindle together form an open quadrilateral without bracing (fig. 315), *a b* and *d c* being the fork sides, *b c* the crank-bracket, and *a d* the wheel spindle. If this structure be acted on by forces there will be in general a tendency to distortion. The tension of the chain, *ef*, is such that the points *e* and *f* on the spindle and crank-bracket respectively, in the plane of the chain, tend to approach each other, and the structure is distorted into the position a^1 *b c* d^1. The action can be easily imagined by supposing the structure jointed at the corners *a*, *b*, *c*, and *d*. In the actual structure this distortion is only resisted by the stiffness of the joints, and the bending-moment can be investigated thus: Consider the equilibrium of the wheel spindle *a d* (fig. 316). It is acted on by the pressure on the two bearings (the resultant of which is the pull of the chain *ef*), and the forces exerted by the fork sides at the points *a* and *d* respectively.

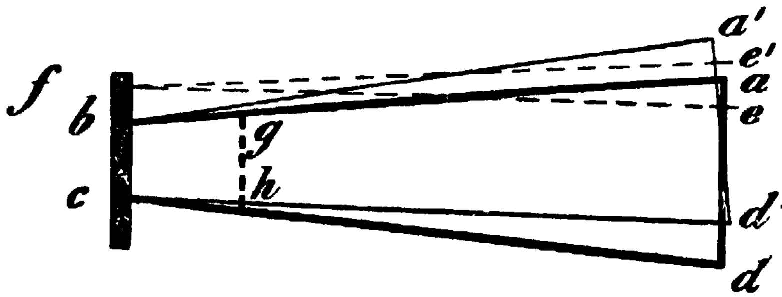


FIG. 315.

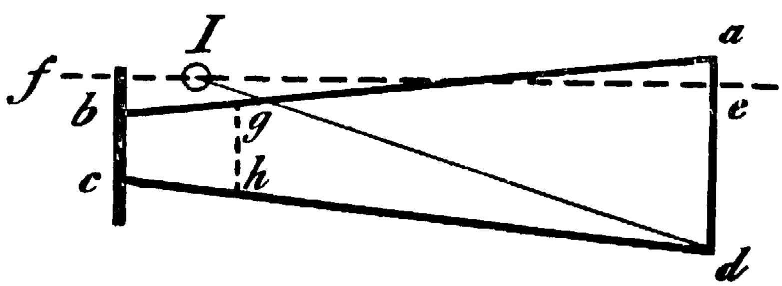


FIG. 316.

The spindle is acted on by three forces, which, being in equilibrium, must all pass through the same point *I*, lying somewhere on the line *e f* produced indefinitely in both directions. Thus, the force acting on the fork side *a b* is in the direction *a I*. If F_2 be the magnitude of this force, and l_2 the perpendicular from *b* on *a I*, there will be a bending-moment $M_2 = F_2 l_2$. With a similar notation for the fork side *c d*, there will be a bending-moment $M_3 = F_3 l_3$ at the point *c* of the fork side. If *I* coincided with the point of intersection of *a b* and *e f*, M_2 would be zero, but M_3 would be very great.

Example III.—We might assume such a position for *I* that M_2

and M_3 would be approximately equal. In this position, taking the data of the above examples, l_2 would be about $\frac{3}{8}$ in., and

$$M_2 = F_2\,l_2 = 410 \times \tfrac{3}{8} = 154 \text{ inch-lbs.}$$

If the lower fork be of round tube $\frac{3}{4}$ in. diameter, 20 W.G., we find, from Table IV., p 113, $Z = \cdot 0137$ in.^{3} Substituting in the formula $M = Zf$ we get

$$f = \frac{154}{\cdot 0137} = 11{,}200 \text{ lbs. per sq. in.}$$

The sectional area of the tube, from Table IV., p. 113, is ·0807 sq. in.; therefore the stress due to the compression of 410 lbs. is

$$f = \frac{410}{\cdot 0807} = 5{,}080 \text{ lbs. per sq. in.}$$

Thus, the maximum compressive stress on the fork at b is

$$f = 11{,}200 + 5{,}080 = 16{,}280 \text{ lbs. per sq. in.}$$

Section of Chain-struts.—The tubes from which the chain-struts are made are usually of round section. Occasionally tubes of oval section are used, the larger diameter of the tube being placed vertically. Since the plane of bending of the fork tubes is horizontal, if the fullest advantage be desired the oval tubes should be placed with the larger diameter horizontal. But the horizontal diameter is limited by the necessity of getting a narrow tread. For a given sectional area (or weight) of tube, and horizontal diameter, the bending resistance will be greater, the greater the vertical diameter and the less the thickness of the tube; since a larger proportion of the material will be at the greatest distance from the neutral axis.

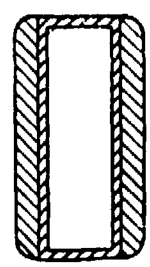


FIG. 317.

D tubes have also been used with the flat side vertical. The discussion in section 98 has shown a difference of about one per cent. in favour of the **D** tube consisting of a semicircle and its diameter. Square or rectangular tubes have not been used to any great extent for the chain-struts, but the discussion in section 99 shows that for equal sectional area and diameter they are much stronger than the round tube. If the

horizontal diameter b be constant, and the vertical unlimited, a rectangular tube with great vertical diameter will be stronger, weight for weight, than a square tube; Z approaching the value $\frac{A\,b}{2}$, corresponding to the whole sectional area being concentrated at the two sides parallel to the neutral axis, the other two sides being indefinitely thin.

A still more economical section for the lower fork tubes would be a hollow rectangle, the vertical sides being longer and thicker than the horizontal. This might be attained by drawing a thin rectangular tube of uniform thickness, and brazing two flat strips on its wider faces (fig. 317).

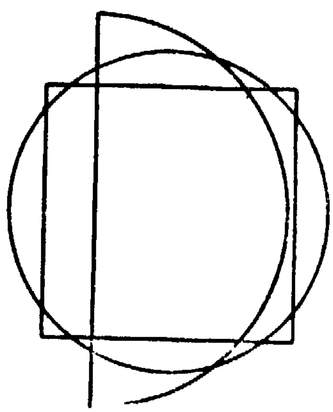


FIG. 318.

Figure 318 shows the sections of round, D, and square tubes of equal perimeter.

Lower Fork with Bridge Bracket.—If the cog-wheel on the crank-axle be placed between the two bearings, as in the 'Ormonde' bicycle (fig. 259), the chain will run between the two lower fork sides (fig. 319), and there will be no bending stresses on the fork tubes due to the pull of the chain. The objection to this arrangement is that the tread must be increased considerably in order to have a bearing outside the cog-wheel on the crank-axle.

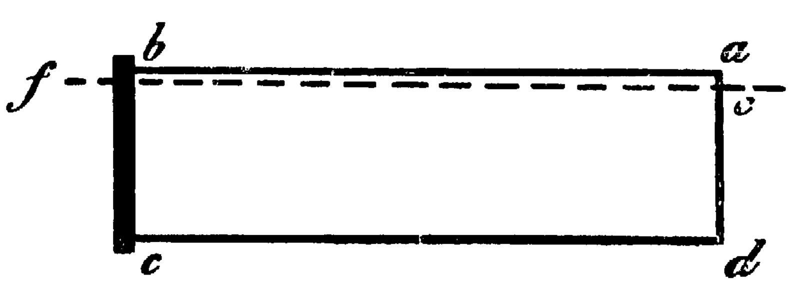


FIG. 319.

'Referee' Lower Back Fork.—In the 'Referee' bicycle the bending on the fork

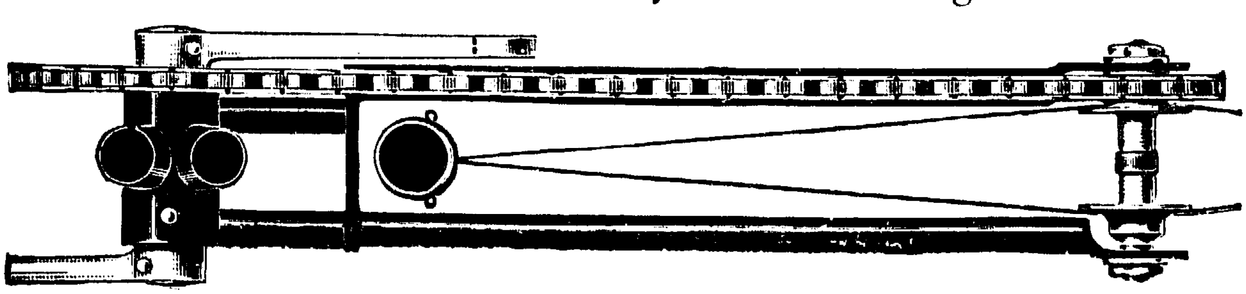


FIG. 320.

sides is eliminated by an ingenious arrangement shown in figure

320. The fork tubes are parallel to the plane of the chain, and instead of running forward to the crank-bracket, they end at an intermediate bridge piece connected to the crank-bracket by two parallel tubes lying closer together than the fork sides. If the end lugs to which the ends of the driving-wheel spindle are fastened were central with the tubes, the bending stresses might be entirely confined to the bridge piece.

239. **Tandem Bicycle Frames.**—The design of tandem frames is much more difficult than that of single bicycle frames, since

FIG. 321.

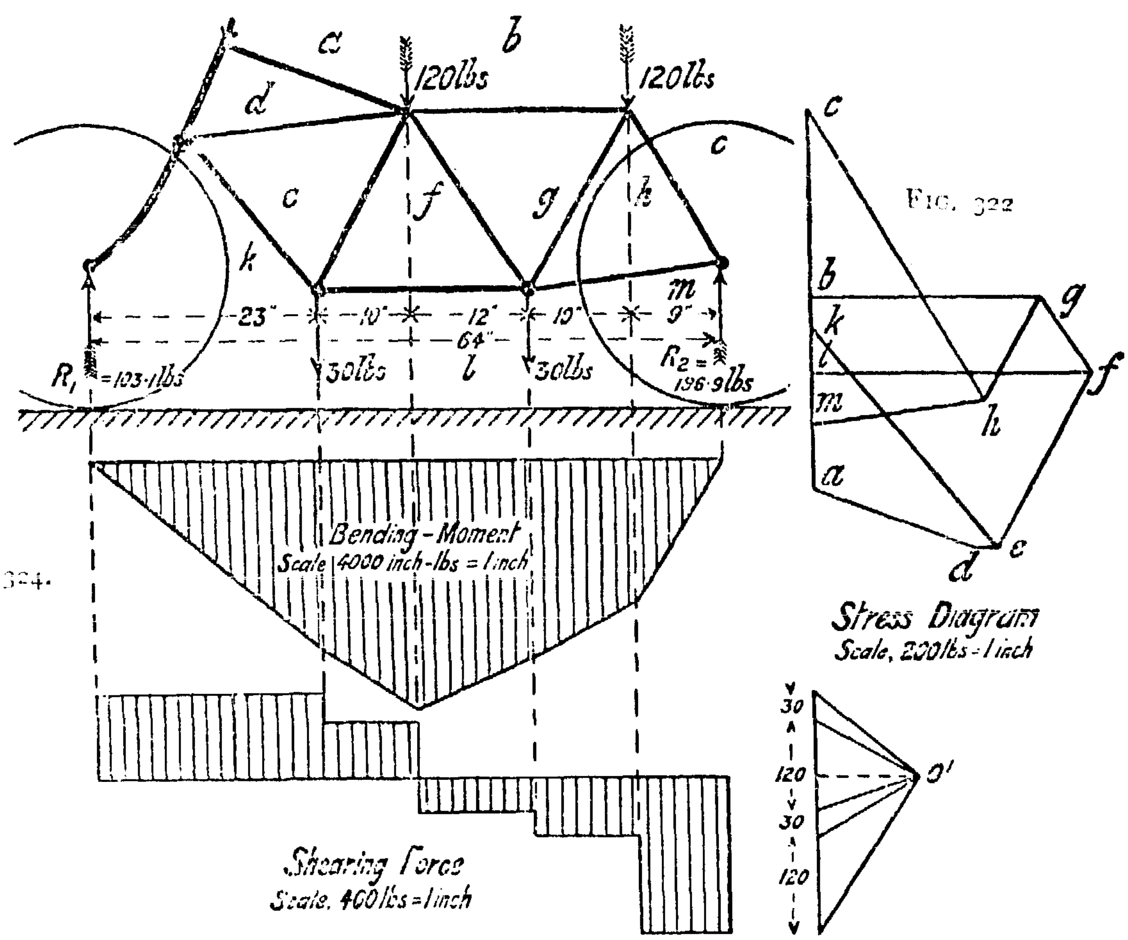


FIG. 322.

FIG. 324.

FIG. 323.

the weight to be carried is double, and the span of the present popular type of tandem from centre to centre of wheels is also greater than that of the single machine. The maximum bending-moment on a tandem frame is therefore much greater than that on the frame of a single bicycle. If, however, one of the riders overhangs the wheel centre, the maximum bending-moment on the frame may actually be less than on that of the single machine.

In the 'Rucker' tandem bicycle (fig. 135) each rider was nearly vertically over the centre of his driving-wheel, and the maximum bending-moment on the backbone was not very great.

Example.—With 120 lbs. applied at the rear saddle, with an overhang of 10 in., the maximum bending-moment was

$$M = 120 \times 10 = 1{,}200 \text{ inch-lbs.}$$

If the maximum stress on the backbone had not to exceed 20,000 lbs.,

$$Z = \frac{1{,}200}{20{,}000} = \cdot 060 \text{ in.}^3$$

A tube $1\frac{1}{4}$ in. diameter, 17 W.G., would have been sufficient.

It may be noticed that with one rider overhanging the wheel-base the bending-moment changes sign about the middle of the frame—*i.e.* if the backbone were originally straight, while carrying the riders the rear portion would be slightly bent with its centre of curvature downwards, the front portion with its centre of curvature upwards.

Figure 321 shows the frame of a rear-driving tandem Safety with both riders between the wheel centres, similar to that of figure 296. The top- and bottom-tubes of the forward portion of the frame should be arranged so that they intersect on the vertical through the front wheel centre, but in order to make the stress-diagram more general they are not so shown in figure 321. Figure 322 shows the stress-diagram, regarding the frame as a plane structure, while figures 323 and 324 are the shearing-force and bending-moment diagrams respectively.

The scale of the stress-diagram, 200 lbs. to an inch, has been chosen half that of the stress-diagram of the single machine (fig. 297), and a few comparisons may be made. The thrusts on the top-tubes, $a\,d$ and $b\,g$, of the tandem are respectively about $2\frac{1}{2}$ and $3\frac{1}{2}$ times that on the top-tube of the single machine. The pull on the front bottom-tube, $e\,k$, of the tandem is about $2\frac{1}{2}$ times that for the single. The thrusts on the diagonal, $f\,g$, and the front down-tube, $e\,f$, are respectively $3\frac{1}{2}$ and $6\frac{1}{2}$ times that on the down-tube of the single machine ; while the *pull* on the rear down-tube, $g\,h$, of the tandem is about four times the thrust on the down-tube of the single machine. The pulls on the lower back fork, $m\,h$, and

on the middle chain-struts, *f l*, are respectively about 2 and $3\frac{1}{2}$ times that on the lower back fork of the single machine.

In making the above comparisons it should be remembered that the single frame (fig. 296) is relatively higher than the tandem frame (fig. 321) illustrated. If the latter were higher, the stresses on its members would be less.

The scale of the bending-moment diagram (fig. 324) is 4,000 inch-lbs. to an inch, twice that for the single machine (fig. 299). The maximum bending-moment is more than three times that for the single machine.

A glance at the shearing-force diagrams (figs. 298 and 323) shows that on a vertical section passing through the rear down-tube of the tandem the shear is negative, while at the down-tube of the single machine the shear is positive. Hence the stress on the rear down-tube is tensile. This can also be shown by a glance at the force-polygon, *l m h g f* (fig. 322), for the five forces acting at the rear crank-bracket (fig. 321); the force *h g*, being directed away from the bracket, indicates a pull on the down-tube.

The thrust on the tube *d e* is small, and vanishes when the front top- and bottom-tubes intersect vertically above the front wheel centre. The thrust on the diagonal tube, *f g*, of the middle parallelogram is 60 lbs., smaller than the thrust or pull on any other member of the frame. This explains why the frame with open parallelogram (fig. 267) and those with no proper diagonal bracing are able to stand for any time under the loads to which they are subjected.

The maximum stresses on the members of the frame due to the vertical loads will be largely increased by the stresses due to the pull of the chain, the thrust of the pedals, and the seat adjustment, as already discussed. The magnitudes of these stresses will be proportionately greater in the tandem than in the single frame.

Tandem frames may be also subjected to considerable twisting strains. If the front and rear riders sit on opposite sides of the central plane of the machine, the middle part will be subjected to torsion. This torsion can be best resisted by one tube of large diameter; *no arrangement of bracing in a plane can strengthen a tandem frame against twisting.*

240. **Stresses on Tricycle Frames.**—Nearly all the frames of early tricycles were unbraced, and their strength depended entirely on the thickness and diameter of the tubes used, one exception being that of the 'Coventry Rotary' (fig. 144), the side portion of which formed practically a triangular truss; another, that of the 'Invincible,' a central portion of which was fairly well braced.

In the early 'Cripper' tricycles the frame was usually of T shape, and consisted of a *bridge* supporting the axle, and a *back-bone* supporting the saddle and crank-axle. With the usual arrangements of wheels and saddle, about three-eighths of the weight of the rider rested on each driving-wheel. The strength of the bridge can easily be calculated thus:

Example I.—If the weight transferred to the middle of the bridge be 120 lbs., the track of the wheels be 30 inches apart, the middle of the bridge is subjected to a bending-moment

$$M = \frac{W\,l}{4} = \frac{120 \times 30}{4} = 900 \text{ inch-lbs.}$$

If the maximum stress be 20,000 lbs. per sq. in.,

$$Z = \frac{M}{f} = \frac{900}{20{,}000} = \cdot 045 \text{ in.}^3$$

A tube $1\frac{1}{8}$ in. diameter, 17 W.G. (see Table IV.), will be sufficient.

In calculating the strength of the backbone the worst case will be when the total weight of the rider is applied at the crank-axle. Taking the relative distances as in the Safety bicycle (fig. 296),

$$M = \frac{150 \times 23 \times 19}{42} = 1{,}560 \text{ inch-lbs.}$$

$$Z = \frac{1{,}560}{20{,}000} = \cdot 078 \text{ in.}^3$$

A tube $1\frac{3}{8}$ in. diameter, 16 W.G., will be sufficient.

With frames made on the same general design as that of the Safety bicycle the stresses will be calculated as already discussed for the bicycle, the only important additional part being the bridge supporting the axle. Its strength may be calculated as in the above example. The stresses on the axle-bridge are diminished by taking the seat-struts to the outer end of the bridge, as in

'Starley's' frame (fig. 153), and in the 'Singer' frame (fig. 273). Figure 325 is plan and elevation of the rear portion of 'Starley's' frame. At the outer end of the bridge, which in this case is a tube concentric with the axle, there are three forces acting, which, however, do not all lie in the same plane. These are the re action of the wheel R, the thrust T along the seat-strut, and the pull A along the bridge. These forces in the plan are denoted by the corresponding small letters, and in the elevation

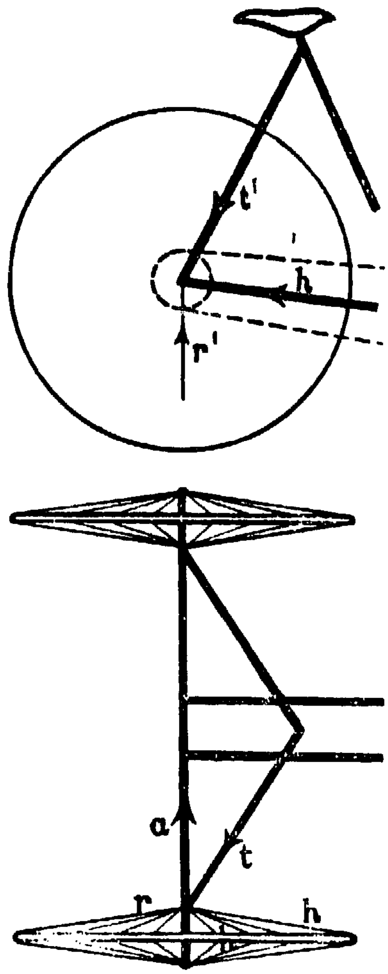


FIG. 325.

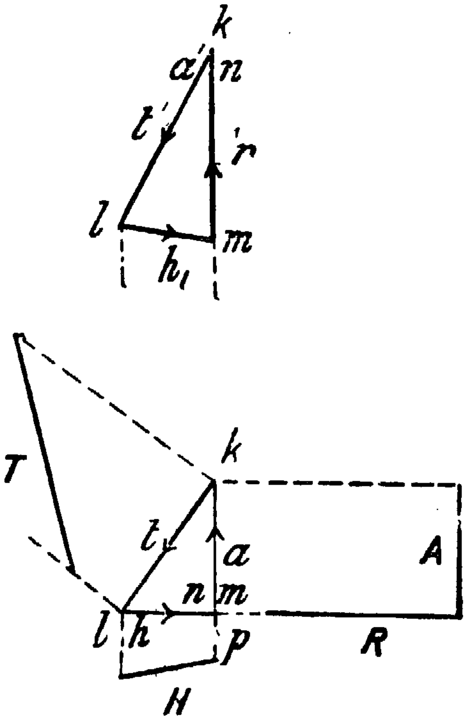


FIG. 326.

with the corresponding small letter with a dash (^{1}) attached. If a force, H, parallel to the chain-struts be applied at the end of the axle, the four forces H, A, R, and T will be in equilibrium, and may be represented by four successive edges of a tetrahedron respectively parallel to the direction of the forces. The plan and elevation of this tetrahedron, $k\ l\ m\ n$, is drawn in figure 326, the length of the side corresponding to the force R being drawn to any convenient scale. The magnitudes of the forces H, A, and

T can be measured off from the true lengths of the corresponding edges of the tetrahedron. These are shown in the plan.

Example II.—Suppose $R = 60$ lbs., and the direction of the tubes is such that $H = 30$ lbs., the resultant of the three forces R A, and T is equal and opposite to H; thus the bridge is subjected to a bending in the plane of the chain-strut. If the distance from the end to the centre of the bridge be 14 in.,

$$M = 30 \times 14 = 420 \text{ inch-lbs.}$$

If the bridge be 1 in. diameter and 20 W.G.,

$$Z = \cdot 0253 \text{ in.}^3, \text{ and } f = \frac{420}{\cdot 0253} = 16{,}600 \text{ lbs. per sq. in.}$$

The axle will also be subjected to a bending-moment in a vertical plane, due to the fact that the centre of the wheel is overhung some distance from the end of the bridge. If the overhang be 3 in., the bending-moment $= 60 \times 3 = 180$ inch-lbs., a smaller value than that found above.

241. **The Front-frame.**—The front-frame (fig. 327) is acted on by three forces—the reaction R_1 of the front wheel on its spindle, and the reactions H_1 and H_2 of the ball-head on the steering-tube. Since the front-frame is in equilibrium under the action of these forces, they must all pass through a point h, situated somewhere on the vertical line passing through the wheel centre. If we assume that the direction of the force H_2 at the upper bearing of the ball-head is at right angles to the head, the point h will be determined; the magnitudes of H_1 and H_2 can then easily be determined by an application of the triangle of forces.

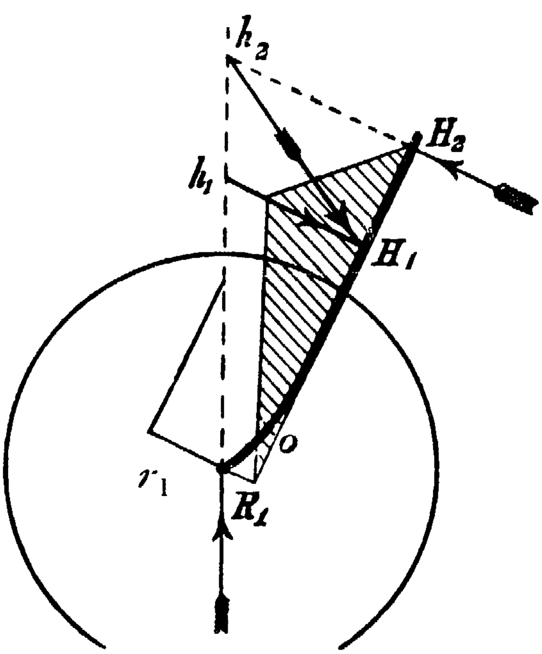


FIG. 327.

Let r_1, h_1, h_2 be the components of the forces R_1, H_1, H_2 at right angles to the ball-head, then the front-frame is subjected to a bending-moment due to these three forces, and the bending-

moment diagram may be represented by the shaded triangle (fig. 327).

Example I.—If $R_1 = 40$ lbs., the slope of the ball head be such that $r = 20$ lbs., and the distance between the lines of action of r_1 and h_1 be 17 in., the greatest bending-moment will be

$$M = 20 \times 17 = 340 \text{ inch-lbs.}$$

If the steering-tube be 1 in. diameter, 20 W.G., we get from Table IV., p. 113, $Z = \cdot 0253$, and the maximum stress on the tube will be

$$f = \frac{M}{Z} = \frac{340}{\cdot 0253} = 13{,}440 \text{ lbs. per sq. in.}$$

It is now becoming usual to strengthen the steering-tube by a liner at its lower end. For the nearest approximation to uniform strength throughout its length it is evident, from the shape of the bending-moment diagram, that its section should vary uniformly from top to bottom. If the liner extend half the length of the ball-head the tube will be of equal strength at the bottom and the middle, and will have an excess of strength at other points.

In a tandem bicycle the nature of the forces on the front-frame are exactly the same as above discussed, but are greater in magnitude. If in a tandem $R_1 = 100$ lbs., with the same dimensions as given above, M will be 850 inch-lbs.

Example II.—If the steering-tube be 1 in. diameter, 18 W.G., and be reinforced by a liner, 18 W.G., the combined thickness of tube and liner is ·096 inches, a little greater than that of a tube 13 W.G. The Z of a 1-in. tube 13 W.G. is ·055, therefore the maximum stress on the tube is

$$f = \frac{850}{\cdot 055} = 15{,}460 \text{ lbs. per sq. in.}$$

The Fork Sides, at their junction to the crown, have to resist nearly the maximum bending-moment (fig. 327). They are usually made of tubes of oval section, tapering towards the wheel centre. The discussion of tubes of oval and rectangular sections (secs. 97 and 99) has shown the latter form to be the superior; and, as there is no limitation of space to be considered in designing the

front fork, the sides may with advantage be made of rectangular tube. If the rectangular tube be of uniform thickness, it has been stated (sec. 99) that for the greatest strength its depth should be three times its width. A still greater economy can be got by thickening the sides of the tube parallel to the neutral axis, either by brazing strips to a tube of uniform thickness (fig. 328), or during the process of drawing.

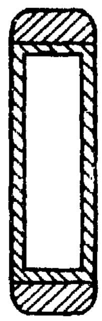


FIG. 328.

Pressure on Crown-plates.—The forces acting on the fork (fig. 329) are R_1, F_1, and F_2, the reactions of the crown-plates.

Example III.—Let the crown-plates be $\frac{3}{4}$ in. apart. Taking the components of these forces at right angles to the steering-head, and taking moments about the centre of the upper plate, we get

$$f_1 = \frac{16{\cdot}5}{{\cdot}75} \times 20 = 440 \text{ lbs., } i.e. \text{ } 220 \text{ lbs. on each side.}$$

In the same way we get

$f_2 = 420$ lbs., *i.e.* 210 lbs. on each side.

The great advantage of the plate crown over the old solid crown is that the forces f_1 and f_2 are made to act as far apart as possible with a given depth of crown, whereas with the older solid crown the pressure was distributed over quite an appreciable distance, so that the distance between the resultant pressures f_1 and f_2 was small; the forces f_1 and f_2 were therefore correspondingly larger, since the moment to be resisted was the same.

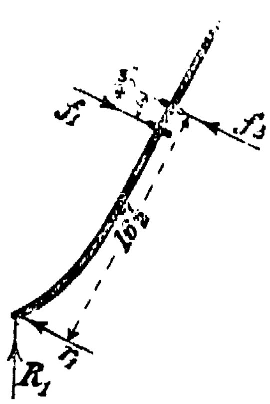


FIG. 329.

In some recent designs of crowns the two plates are united by short tubes *outside* of the fork sides. As regards the attachment of the fork sides, this arrangement is therefore practically equivalent to the old solid crown. If any strengthening is desired, it should be done by an inside liner. Triple crown-plates have been used for tandems; but, as far as we can see, the middle plate contributes nothing to the strength of the joint, and may with advantage be omitted

Handle bar.—The handle-bar, when pulled upwards by the

rider with a force P at each handle, is subjected to a bending-moment Pl, l being the distance from the handle to the centre of the handle-pillar.

Example IV.—If l = 12 in., then M = 12 P inch-lbs. Let the handle-bar be $\frac{7}{8}$ in. diameter, 18 W.G., Z = ·0244 in.^{3}, and let the maximum stress on the handle-bar, f, be 20,000 lbs. per sq. in.; substituting in the formula $M = Zf$, we get

$$12\ P = \cdot 0244 \times 20{,}000.$$
$$\therefore \quad P = 41 \text{ lbs.}$$

That is, a total upward pull of 82 lbs. will produce a maximum stress of 20,000 lbs. per sq. in.

If the handles be bent backwards, the handle-bar is also subjected to a twisting-moment, which, however, usually produces smaller stresses than the bending-moment. For example, if the handle be bent 4 in. backwards, the twisting-moment $T = 4\ P$. The modulus of resistance to torsion of a $\frac{7}{8}$ in. tube, 18 W.G., is, from Table IV., p. 113, ·0488 in.^{3}; and therefore, with the same value for P as in the above example, we get $4 \times 41 = \cdot 0488 f$,

or $$f = 3{,}360 \text{ lbs. per sq. in.}$$

242. **General Considerations Relating to Design of Frame.**—The importance of having the forces acting on a tie or strut exactly central cannot be over-estimated; the few examples already given above show how the maximum stress is enormously increased by a very slight deviation of the applied force from the axis. In iron bridge or roof building, this point is thoroughly appreciated by engineers; but in bicycle building the forces acting on each tube of a frame are, as a rule, so small that tubes of the smallest section theoretically possible cannot be conveniently made. The tubes on the market are so much greater in sectional area than those of minimum theoretical section that they are strong enough to resist the increased stresses due to eccentricity of application of the forces; and thus little or no attention has been paid to this important point of design.

The consideration of the shearing-force and bending-moment diagrams simultaneously with the outline of the frame is instructive, and reveals at a glance some weak points in various types of

frames. The vertical section at any point of a properly braced frame will cut three members; the moment of the horizontal components of the forces acting on these members will be equal to the bending-moment at the section, while the sum of the vertical components will be equal to the shearing-force. Therefore, in general, any part of a frame in which the vertical depth is small will be a place of weakness. The Ladies' Safety frames (figs. 264 and 265) have already been discussed. That shown in figure 266 is weakest at the point of crossing of the two tubes to the steering-head, the depth of the frame being zero at this point, so that only the *bending* resistance of these tubes can be relied on. The cross-frame (fig. 249) is very weak in the backbone, just behind the point where the down-tube crosses it. The Sparkbrook frame (fig. 253) is weakest at a point on the top-tube, just in front of the point of attachment of the tube from the crank-bracket. The frames (figs. 244 and 245) are practically equivalent to a single tube unbraced. The frames shown in figures 247–251 are weakest just behind the steering-head.

The consideration of the shearing-force curve shows the necessity for the provision of the diagonal of the central parallelogram in a tandem frame. The top- and bottom-tubes are nearly horizontal, so that if they were acted on by forces parallel to their axes they could not resist the shearing-force. The shearing-force must therefore be resisted by an inclined member of the frame, or, failing this, the forces on the top- and bottom-tubes cannot be parallel to their axes, and they must be subjected to bending. The same remarks apply to a frame formed by the duplication of either the top- or bottom-tubes without the provision of a diagonal, as in figure 268.

CHAPTER XXIV

WHEELS

243. **Introductory.**—Wheels may be divided into two classes—rolling wheels and non-rolling wheels. In rolling wheels the instantaneous axis of rotation is at the circumference; examples are, bicycle wheels, vehicle wheels, railway carriage wheels, &c. Such rolling wheels have, in general, a *fixed* axis of rotation relative to the frame, which has a motion of translation when the wheel rolls. Non-rolling wheels are those not included in the above class; they may be mounted on fixed axes, their circumferences being free, or in contact with other wheels. Such are fly-wheels, gear-wheels, rope- or belt-pulleys, &c.

Wheels may again be subdivided, from a structural point of view, into solid wheels, wheels with arms, nave, and rim, cast or stamped in one piece, and built-up wheels. In a solid rolling wheel, the load applied at the centre of the wheel is transmitted by compression of the material of the wheel to its point of contact with the ground.

244. **Compression-spoke Wheels.**—A built-up wheel usually consists of three portions—the *hub* (nave, or boss), at the centre of the wheel; the *rim* or *periphery* of the wheel; and the *spokes* or *arms*, connecting the rim to the hub. Built-up wheels may be divided again into two classes, according to the method of action of the spokes. A wheel may be conceived to be made without a rim, consisting only of nave and spokes (fig. 330). In this case the load applied at the centre of the wheel is evidently transmitted by compression of the spoke, which is at the instant in contact with the ground. If the spokes are numerous, the rolling motion over a hard surface may be made fairly regular. In the ordinary

wooden cart or carriage wheel (fig. 331), the ends of the spokes are connected by wooden felloes, *f*, the felloes being mortised to receive the spoke ends, and an iron tyre, *t*, encircles the whole. This iron tyre is usually shrunk on when hot, and in cooling it compresses the felloes and spokes. This construction is very simple, since only one piece—the iron tyre—is required

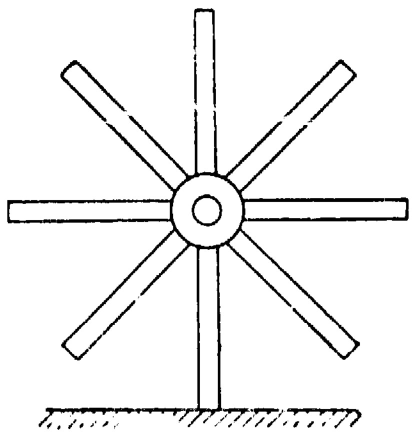


FIG. 330.

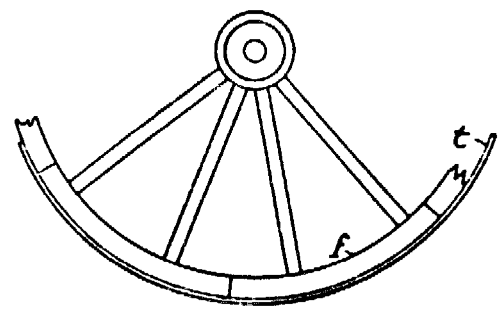


FIG. 331.

to bind the whole structure together. The compression wheel compares favourably in this respect with the tension wheel. On the other hand, the sectional area of the spokes must be great, in order to resist buckling under the compression; very light wheels cannot, therefore, be made with compression spokes. The method of transmitting the load from the centre of the wheel to the ground is practically the same as in figure 330.

245. **Tension-spoke Wheels.**—The initial stresses in a bicycle wheel of the usual construction are exactly the reverse of those

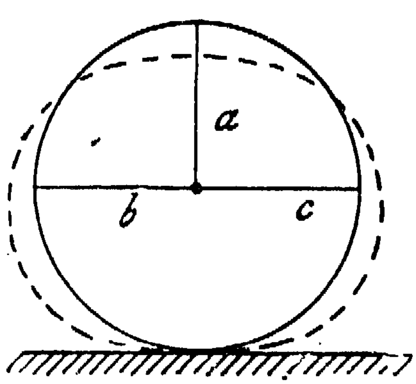


FIG. 332.

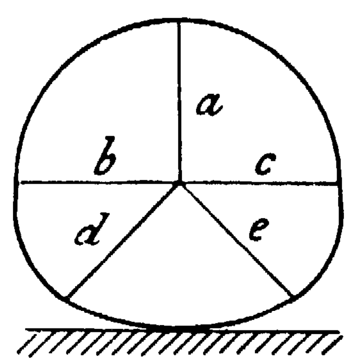


FIG. 333.

on the compression-spoke wheel (fig. 331). The method of action of the tension-spoke wheel may be shown as follows. Suppose the hub connected by a single wire, *a*, to a point on the top of the

rim, a load applied at the centre of the wheel would be transferred to the top of the rim and would tend to flatten it, the sides would tend to bulge outwards, and the rim to assume the shape shown by the dotted lines (fig. 332). This horizontal bulging might be prevented by connecting the hub to the rim by two additional spokes, *b* and *c*. If, now, a load were applied at the centre of the wheel, the three spokes, *a*, *b*, and *c*, would be subjected to tension, and if the rim were not very stiff it would tend to flatten at its lower part, as indicated in figure 333. Additional spokes, *d* and *e*, would restrain this bulging. Thus, by using a sufficient number of spokes capable of resisting tension, the load applied at the centre of the wheel can be transmitted to the ground without appreciable distortion of the rim.

246. **Initial Compression in Rim.**—In building a bicycle-wheel the spokes are always screwed up until they are fairly tight.

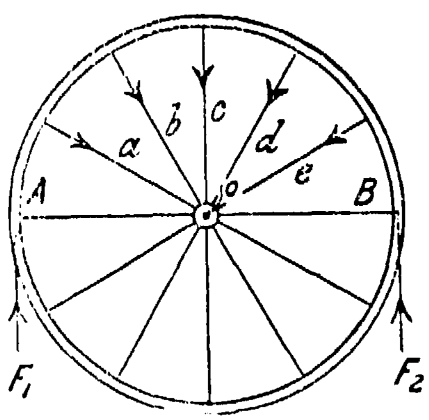


FIG. 334.

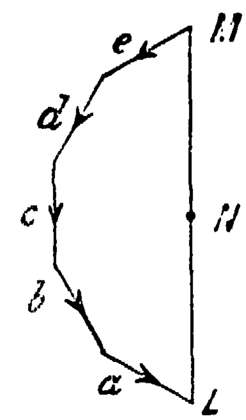


FIG. 335

The tension on all the spokes should be, of course, the same. This tightening up of the spokes will throw an initial compression on the rim, which may be determined as follows. Suppose the rim cut by a plane, *A O B*, passing through the centre of the wheel (fig. 334). Consider the equilibrium of the upper portion of the rim of the wheel : it is acted on by the pulls of the spokes *a*, *b*, *c*, *d* . . . and the reactions F_1 and F_2 of the lower part of the rim at *A* and *B*. If the tension *t* be the same in all the spokes, the force-polygon *a*, *b*, *c*, *d* . . . (fig. 335) will be half of a regular polygon. The sum of the forces *F* at *A* and *B* will be equal to the closing side, *L M*, of the force-polygon.

If the number of spokes in the wheel be great, the force-polygon (fig. 335) may be considered a circle. Then, if *n* be the

number of spokes in the wheel, the circumference LM (fig. 335) is equal to $\frac{nt}{2}$, the diameter LM to $\frac{nt}{\pi}$.

But $$2F = LM = \frac{nt}{\pi};$$

therefore $$F = \frac{nt}{2\pi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Example.—The driving-wheel of a Safety has 40 spokes, No. 14 W.G., which are screwed up to a tension of 10,000 lbs. per sq. in. Find the compression on the rim.

From Table XII., page 346, the sectional area of each spoke is ·00503 sq. in.; the pull t is therefore $\cdot 00503 \times 10{,}000 = 50\cdot 3$ lbs. Substituting in (1),

$$F = \frac{40 \times 50\cdot 3}{2 \times 3\cdot 1416} = 320 \text{ lbs.}$$

247. **Direct-spoke Driving-wheel.**—The mode of transmission of the load from the centre of a bicycle wheel to the ground having been explained, it remains to show how the driving effort is transmitted from the hub to the rim. In a large gear-wheel the arms are rigidly fixed to the nave, and while a driving effort is being exerted, the arms press on the rim of the wheel in a tangential direction. Thus each arm may be considered as a beam rigidly fixed to the nave and loaded by a force at its end near the rim. The spokes of a bicycle wheel are not stiff enough to transmit in this manner forces transverse to their axis, being to all intents and purposes perfectly flexible. When a driving force is exerted the hub turns through a small angle without moving the rim, so that the spokes whose axes initially all passed through the centre of the wheel now touch a circle, s (fig. 336). Let r be the radius of this circle, and P the pull of the driving chain which is exerted at a radius R. Considering the equilibrium of the hub, the moment of the force P about the centre is PR;

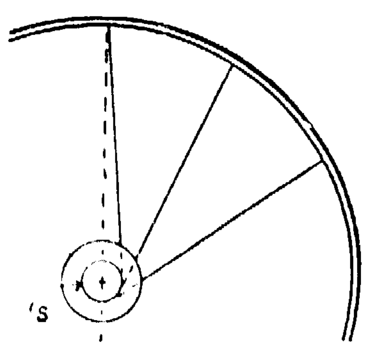


Fig. 336.

the moment of the forces due to the pull of the spokes on the hub is

$$n\,t\,r.$$

Thus, $PR = n\,t\,r$,

and $$r = \frac{PR}{n\,t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

Example.—Let the driving-wheel have 40 spokes, each with an initial tension of 50 lbs.; let the pull of the chain be 300 lbs., and be exerted at a radius of $1\frac{1}{2}$ in. Find the size of the circle s, and the angle of displacement of the hub.

Substituting in (2)

$$r = \frac{300 \times 1\frac{1}{2}}{40 \times 50} = \cdot 225''.$$

Figure 337 is a drawing showing the displacement of the hub. Let $c\,d$ be the radius of the circle touched by the spokes, $b\,a$ the initial position of a spoke, $b^1\,a^1$ the displaced position, and let the distance of the point of attachment of the spokes from the centre of the hub be $\frac{7}{8}$ in.; the angle of displacement of the hub, $a\,c\,a^1$ will be approximately

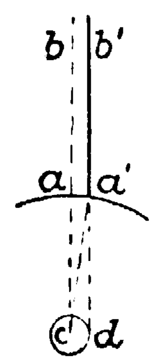


FIG. 337.

$$\frac{\cdot 225}{\cdot 875} = \cdot 257 \text{ radians,}$$

or, $$\frac{\cdot 257 \times 180}{\pi} = 14\cdot 7 \text{ deg.}$$

If the driving effort be reversed, as in back-pedalling, the hub will first return to its original position relative to the rim, and then be displaced in the opposite direction before the reversed driving effort can be transmitted.

Thus, a direct-spoke bicycle wheel is not a rigid structure, but has quite a perceptible amount of tangential flexibility between the hub and the rim.

Lever Tension Driving-wheels.—In the early days of the 'Ordinary,' wheels were often made with a pair of long levers projecting from the hub, from the ends of which wires went off to the rim. These tangential wires were adjustable, and by tightening them the rim was moved round relative to the

hub, and thus the tension on the spokes could be adjusted. The tangential driving effort was also supposed to be transferred from the hub to the rim by the lever and tangent wires, while the

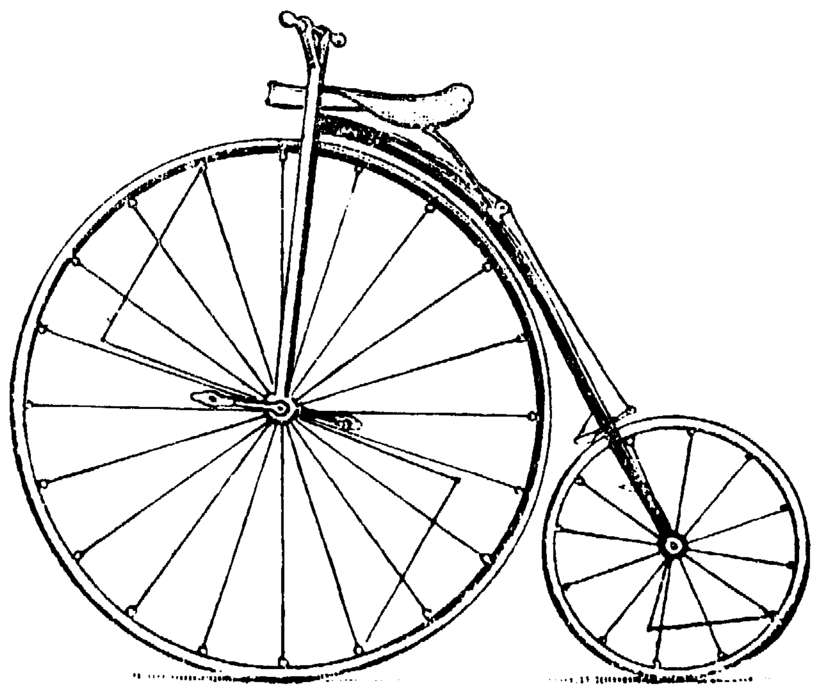


FIG. 338.

radial spokes only transmitted the weight from the hub to the rim. Figure 338 shows the 'Ariel' bicycle with a pair of lever tension wheels.

248. **Tangent-spoke Wheels.**—In a tangent wheel the spokes are not arranged radially, but touch a circle concentric with the hub (fig. 339). The pull on the tangent-spokes indicated by the full lines would tend to make the hub turn in the direction of the arrow. Another set of spokes, represented by the dotted lines, must be laid inclined in the opposite direction, so that the hub may be in equilibrium. The initial tension should be the same on all the spokes.

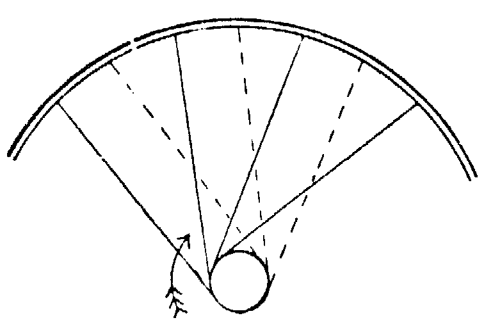


FIG. 339.

Let a driving effort in the direction of the arrow be applied at the hub. This will have the effect of increasing the tension on one half of the spokes and diminishing the tension on the other half. If r be the radius of the circle to which the spokes are

tangential, t_1 and t_2 the tensions on the tight and slack spokes respectively, the total tangential pull of the spokes at the hub is

$$\frac{n}{2}(t_1 - t_2).$$

Therefore

$$PR = \frac{nr}{2}(t_1 - t_2),$$

from which

$$t_1 - t_2 = \frac{2PR}{nr} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Example.—Let r be $\frac{7}{8}$ in., the spokes 15 W.G., the modulus of elasticity of the spokes 10,000 tons per sq. in.; then, taking the rest of the data as in section 247, find the angle of displacement of the hub relative to the rim under the driving effort.

Substituting in (3),

$$t_1 - t_2 = \frac{2 \times 300 \times 1\frac{1}{2}}{40 \times \frac{7}{8}} = 25{\cdot}7 \text{ lbs.}$$

The sectional area of each spoke (Table XII.) is ·00407 sq. in.; the increase or diminution of the tension due to the pull of the chain is therefore

$$\frac{25{\cdot}7}{2 \times {\cdot}00407} = 3{,}156 \text{ lbs. per sq. in.} = 1{\cdot}41 \text{ tons per sq. in.}$$

The extension of one set of spokes and the contraction of the other set will thus be $\frac{1{\cdot}41}{10{,}000}$ th part of their original length, which length in a 28-in. driving-wheel is about 12 in. The displacement of a point on the circle of radius $\frac{7}{8}$ in. is thus

$$\frac{1{\cdot}41 \times 12}{10{,}000} = {\cdot}00169 \text{ in.}$$

The angle the hub is displaced relative to the rim will be

$$\frac{{\cdot}00169 \times 180}{\frac{7}{8} \times \pi} = {\cdot}11 \text{ deg.}$$

Comparing this example with that of section 247, the superiority of the tangent wheel in tangential stiffness is apparent. In this example it should be noted that the initial pull on the spokes does not enter into the calculation. Consequently, the initial pull on tangent-spokes may with advantage be less than that on direct-spokes.

249. **Direct-spokes.**—The spokes of a direct-spoke wheel are usually of the form shown in figure 340, the conical head at the end engaging in the rim, and the other end being screwed into the hub. For the sake of preserving the spoke of equal strength throughout, its end is often *butted* before being screwed (fig. 341), the section at the bottom of the thread in this case being at least as great as at the middle of the spoke.

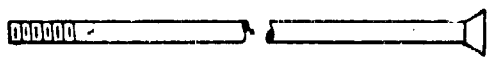


FIG. 340.

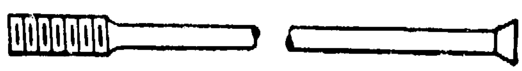


FIG. 341.

In figure 339 the spoke is shown making an acute angle with the hub. As a matter of fact, under the action of a driving effort the spokes near the hub will be bent, as shown exaggerated in figure 342. The continual flexure under the driving effort weakens and ultimately causes breakage of direct spokes, unless made of greater sectional area than would be necessary if they could be connected to the hub by some form of pin-joint. The conical head lies loosely in the rim, and being quite free to adjust itself to any alteration of direction, the spoke near the rim is not subjected to such severe straining actions as at the hub.

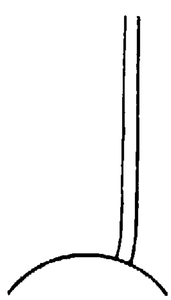


FIG. 342.

250. **Tangent-spokes.** — Tangent-spokes cannot be conveniently screwed into the hub, but are threaded through holes in a flange of the hub, the end of the spoke being made as indicated in figure 343. This sharp bend of the spoke seriously affects its strength. Let P be the pull on the spoke, and d its diameter. On the section of the spoke at a there will be a bending-moment Px, x being the distance between the middle of the

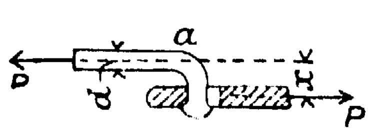


FIG. 343.

section a and the hub flange; this distance may be taken approximately equal to d. The bending-moment is then $P\,d$, $Z = \frac{\pi d^3}{32}$, and the maximum stress, f, due to bending will be found by substitution in the formula $M = Z f$. Therefore

$$P\,d = \frac{\pi d^3}{32} f$$

and

$$f = \frac{32\,P\,d}{\pi d^3} = \frac{32\,P}{\pi d^2}.$$

The tensile stress on the middle of the spoke is

$$\frac{4\,P}{\pi d^2}$$

Thus the stress due to bending on the section at the corner is eight times that on the body of the spoke due to a straight pull.

Figure 344 shows a tangent-spoke strengthened at the end by butting.

The ends of tangent-spokes must be fastened to the rim by means of nuts or nipples. The nipple has its inner surface screwed to fit the screw on the end of the spoke, has a conical head which lies in a corresponding counter-sunk hole in the rim, and a square or hexagonal body threaded through the hole in the rim for screwing up by means of a small spanner.

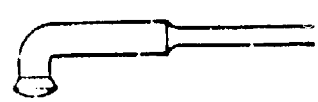


FIG. 344.

A piece of wire threaded through the hub flange (fig. 345), and its ends fastened to the rim by nipples in the usual way, is often used to form a pair of tangent spokes. The objection to the spoke shown in figure 343 still holds with regard to this form; but the fact that no head has to be formed at the hub probably makes it slightly stronger than a single spoke of the same diameter headed at the end.

FIG. 345.

Figure 346 shows the form of tangent-spoke used by the St. George's Engineering Co. in the 'Rapid' cycle wheels. The spoke is quite straight from end to end, and is fastened to the rim in the usual way by a nipple. It is fastened to the hub by means of a short stud projecting from the hub flange, a small hole being drilled in the projecting head of the stud, and the spoke threaded through it. The headed end of the spoke is pulled up against the stud. Spokes of this form are not subjected to bending, and are therefore much stronger than tangent-spokes of the usual form of the same gauge.

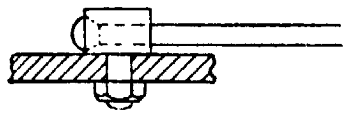


FIG. 346.

TABLE XII.—SECTIONAL AREAS AND WEIGHTS PER 100 FT. LENGTH OF STEEL SPOKES.

Imperial standard wire gauge	Diameter	Sectional area	Weight of 100 ft.
	In.	Sq. in.	Lbs.
6	·192	·02895	10·005
7	·176	·02433	8·409
8	·160	·02011	6·950
9	·144	·01629	5·629
10	·128	·01287	4·447
11	·116	·01057	3·652
12	·104	·00849	2·936
13	·092	·00665	2·298
14	·080	·00503	1·738
15	·072	·00407	1·407
16	·064	·00322	1·112
17	·056	·00246	·850
18	·048	·00181	·625
19	·040	·00126	·434
20	·036	·00102	·352

251. **Sharp's Tangent Wheel.**—The distinctive features of this wheel, invented by the author, are illustrated in figure 347. The hub is suspended from the rim by a series of wire loops, one loop forming a pair of spokes. In figure 347, for the sake of clearness of illustration, one loop or pair of spokes is shown thickened. The ends are fastened to the rim by nuts or nipples in the usual way. There is no fastening of the spokes to the hub,

beyond that due to friction. Figure 348 represents the appearance of the spokes in contact with the hub. The arc of contact of the spoke and hub is a spiral, so that all the ends of the spokes on one side of the middle plane of the wheel begin contact with the hub at the same distance from the middle, the other ends all leaving the hub nearer the middle plane. A wheel could be made with loops of wire having circular contact with the hub, but it would not be symmetrical, and the spokes would not all be of

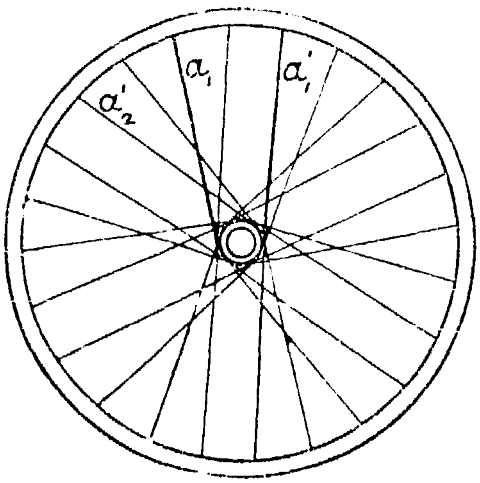


FIG. 347.

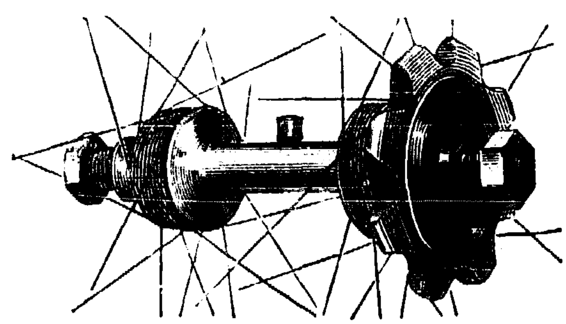


FIG. 348.

the same length. By making the spokes have a spiral arc of contact with the hub, the positions of all the spokes relative to the hub are exactly similar, the wheel is symmetrical, and the spokes are all of the same length. It will be noticed that there are no sudden bends in the spokes, so that they are much stronger than in the ordinary tangent wheel, no additional bending stresses being introduced. For non-driving cycle wheels there can be no question as to the sufficiency of the hub fastening, but it may at first sight seem startling that the mere friction of the spokes on the hub should be sufficient to transmit the driving effort to the rim, though it is well known that by coiling a rope round a smooth drum almost any amount of friction can be obtained. This system of construction is applicable to all types of built-up metal wheels, and has been applied with success to fly-wheels and belt-pulleys, and to the 'Biggest Wheel on Earth'—the gigantic pleasure-wheel at Earl's Court.

Let t be the initial tension on the spokes; then, while the driving effort is being exerted, the tension on one half of each loop

rises to t_1, and on the other half falls to t_2. If t_1 be very much greater than t_2 there will not be sufficient friction between the hub and the wire, and slipping will occur. Let θ be the angle of contact (fig. 349) and μ the coefficient of friction between the spoke and hub. Then, when slipping takes place,

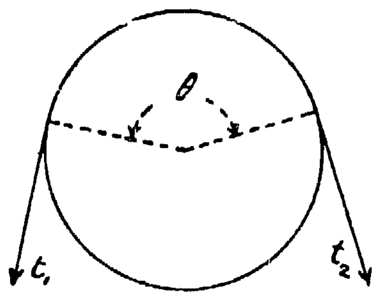


FIG. 349.

$$\frac{t_1}{t_2} = \epsilon^{\mu\theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

If $\frac{t_1}{t_2}$ is less than determined by (4), slipping will not occur.

Equation (4) may be written in the form,

$$log_\epsilon \frac{t_1}{t_2} = \mu\, \theta,$$

the symbol $log_\epsilon \frac{t_1}{t_2}$ denoting the logarithm, to the 'Naperian' or natural base, of the number $\frac{t_1}{t_2}$. Using a table of 'common' logarithms, a more convenient form is—

$$log \frac{t_1}{t_2} = {\cdot}4343\, \mu\, \theta \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Example I.—A driving-wheel 28 in. diameter, on this system, has 40 spokes wrapped round a cylindrical portion of the hub $1\frac{1}{2}$ in. diameter, the initial tension on each spoke is 60 lbs., the pull on the chain is 300 lbs., and is exerted at a radius of $1\frac{1}{2}$ in. Find whether slipping will take place or not.

Let the arc of contact be half a turn, as shown approximately in figure 347, then $\theta = \pi$, the coefficient of friction μ for metal on metal dry surface will be about from ·2 to ·35, but assuming that oil from the bearing may get between the surfaces, we may take a low value, say 0·15 ; substituting in (5)

$$log \frac{t_1}{t_2} = {\cdot}4343 \times {\cdot}15 \times 3{\cdot}1416 = {\cdot}2046,$$

from which, consulting a table of logarithms,

$$\frac{t_1}{t_2} = 1{\cdot}602$$

when slipping takes place. But from (3)

$$t_1 - t_2 = \frac{2\,P\,R}{n\,r} = \frac{2 \times 300 \times 1\frac{1}{2}}{40 \times \frac{3}{4}} = 30 \text{ lbs.}$$

Therefore $$t_1 = 60 + 15 = 75$$
$$t_2 = 60 - 15 = 45$$

and $$\frac{t_1}{t_2} = 1{\cdot}5.$$

Thus, with the above conditions, slipping will not occur.

As a matter of experiment, the author finds that with such a smooth hub and an arc of contact of half a turn slipping takes place in riding up steep hills only when the spokes are initially slacker than is usual in ordinary tangent wheels.

Arc of Contact between Spokes and Hub.—The pair of spokes (fig. 347) is shown having an arc of contact with the hub of nearly two right angles. The arc of contact may be varied. For example, keeping the end a_1 fixed, the other end of the spoke may be moved from $a^1{}_1$ to $a^1{}_2$, or even further, so that the arc of contact may be as shown in figure 350. In this case there are five spoke ends left between the ends of one pair. In general, $4\,n + 1$ spokes must be left between the ends of the same pair, n being an integer.

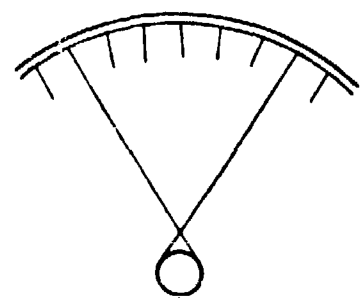


FIG. 350.

In this wheel, should one of the spokes break, a whole loop of wire must be removed. Of course the tendency to break is, as already shown, far less than in direct or tangent spokes of the usual type. If the arc of contact, however, is as shown in figure 347, and a pair of spokes are removed from the wheel, a great additional tension will be thrown on the spoke between the two vacant spaces. If the angle of contact shown in figure 350 be adopted, there will still remain five spokes between the two vacant spaces, so that the additional tension thrown on any single spoke will not be abnormally great.

Grooved Hubs.—The hub surface in contact with the spokes may be left quite smooth, with merely a small flange to preserve the spread of the spokes. The parts of the spokes wrapped round